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REPORT
RAC-R-26

OCTOBER 1967

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RESEARCH ANALYSIS CORPORATION

AD 321500

Engine-Transmission Power Plants for Tactical Vehicles

by
Emil M. Szten, *Project Manager*
Harold G. Seeger
Kenneth R. Simmons
William O. Sprang
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RAC

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SCIENCE AND ENGINEERING DEPARTMENT
RAC REPORT R-26
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RESEARCH ANALYSIS CORPORATION
McLEAN, VIRGINIA

DEPARTMENT OF THE ARMY
OFFICE OF THE CHIEF OF RESEARCH AND DEVELOPMENT
WASHINGTON, D.C. 20310

CRDHF

20 March 1967

SUBJECT: RAC-R-26, "Engine-Transmission Power Plants for Tactical Vehicles"

TO:

1. Transmitted herewith is (are) _____ copy (copies) of RAC-R-26.

2. This document was produced by the Research Analysis Corporation, McLean, Virginia, in conjunction with the contract study of the same title.

3. RAC-R-26 documents an investigation of future engines and power trains and their potential application to tactical vehicles. This report is considered to be a reference document only. The study, its conclusions, and recommendations represent the views of the Research Analysis Corporation, and the document is not an official Department of the Army publication.

FOR THE CHIEF OF RESEARCH AND DEVELOPMENT:

Robert B. Bennett

1 Incl
as

ROBERT B. BENNETT
Colonel, GS
Chief, Human Factors and
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FOREWORD

Technological opportunities for improving the performance of vehicle power plants have increased over the last few years to the point that competition for research and development (R&D) funds required to see each of these opportunities to fruition presents a serious planning problem. A rational analysis of the technological opportunities for the Army's vehicles in terms of operational use and force planning is fundamental to thorough planning for the effective use of limited R&D funds. The use of vehicles in support of ground combat is functionally complex and is not understood sufficiently well to allow technological improvements to be related comprehensively and explicitly to the effectiveness of vehicles in support of combat. This difficulty is propagated into the further problem of distinguishing between the effectiveness of different aggregated mixes of ground vehicles, some technologically advanced and others conservatively engineered, as a basis for force planning.

Much-needed progress toward a practical but thorough and systematic foundation for R&D programs and force planning for vehicles must come from analysis of the functional performance of vehicles in ground-combat missions and accompanying analyses of the relation of this functional performance to engineering characteristics. At the same time such analyses should be supported by a comprehensive survey and technical evaluation of technological advances toward the improved engineering of Army vehicles. The present report provides a survey and technical evaluation, independent of Army and industry viewpoints, of technological opportunities for engines and transmissions for Army ground vehicles operating in the period 1975–1985. Many of these opportunities have evolved in an ad hoc manner in response both to the military and to the civilian markets. The evaluation was made with respect to engineering characteristics and production feasibility.

The study was conducted at RAC in 1966 by a group of vehicle engineering specialists with industrial experience specifically related to the design, test, and manufacture of ground vehicles. The material provided in the report should contribute to future more complete evaluation of vehicle R&D programs using terms of reference that should be formulated from operational and force planning considerations.

Clive G. Whittenbury
Head, Science and Engineering Department

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SUMMARY

Problem

To identify those energy- and power-conversion devices that, if developed and made available, have the greatest potential for advancing the state of the art of tactical vehicles fielded in the 1975–1980 and 1980–1985 time frames, thus providing a base from which priorities for energy- and power-conversion-device development can be established.

Facts

This study was performed for, and at the request of, the Chief of Research and Development, Department of the Army, by RAC.

The results of this study are intended to assist the US Army in selecting those programs for research and development (R&D) that appear to offer the greatest return for investment. Although analysis and discussion of components and systems include those envisioned as feasible within a broad time frame, only devices available for application in vehicles to be fielded in the 1975–1980 and 1980–1985 time frames are emphasized in this document.

Need for the Study

The need for this study was predicated on the fact that the Department of the Army needs up-to-date circumspect guidelines in order to establish priorities for energy- and power-conversion devices required for tactical vehicles for the 1975–1980 and the 1980–1985 time frames. The most recent related study on vehicle power systems was completed in 1956 and is not appropriate for application to the time frames cited.

Vehicle improvement cannot be accomplished without the continued development of energy- and power-conversion devices well in advance of vehicle development. Past experience has shown that on the average 5 years are required to develop a new energy- or power-conversion device and that an additional 5 years are then required to develop a new vehicle.

The study is based on the premise that superior or improved vehicles can be developed if technological advances are obtained in energy- and power-conversion devices. The criteria considered as contributing most to the technical advancement of these devices are:

(a) Reduction in size
(b) Reduction in weight

(c) Increased performance
(d) Increased reliability and decreased maintenance
(e) Reduced initial cost
(f) Reduced operating cost
(g) Compatibility with US Army fuel policy
(h) Vehicle design flexibility
(i) Industry's mobilization capacity
(j) Reduction of proprietary components

Discussion

Assumptions

The study was based on the following assumptions:

(a) The US Army intends to develop energy- and power-conversion devices and supporting components for tactical vehicles. The devices and components designed for military application cannot be obtained on the commercial market.

(b) Energy- and power-conversion systems applicable to aircraft or to stationary equipment are excluded from this study unless information about them is pertinent and has direct applicability to tactical vehicles.

(c) Statements by industry will be accepted only if supported by test data or if the concepts are deemed feasible.

Data Sources

Up-to-date information on energy- and power-conversion devices, related components, and systems has been sought from cognizant and concerned personnel in Federal agencies, the military, industry, and academic and industrial research institutions. Literature sources have included papers published by professional societies and organizations, commercial books, periodicals, Government and industrially sponsored technical reports, and related industrial pamphlets and brochures.

Approach

From the sources given, data on present energy- and power-conversion devices, related systems, and components were obtained and analyzed. The analysis was performed to assess the status of present hardware technology, present R&D programs and concepts, and those research programs that are anticipated.

Four types of tactical vehicles (wheeled, tracked, amphibious, and special-purpose), of the horsepower and weight classes shown in the accompanying tabulation, were considered in the course of this study.

Horsepower	Vehicle weight, tons
<120	<5
120–250	5–15
250–500	15–35
500–1000	35–50
>1000	>50

The study is divided into four parts. The title and contents of each part are as follows:

Part I: Energy-Conversion Devices. The 14 major types of energy-conversion devices are named, analyzed, evaluated, and discussed.

Part II: Power-Conversion Devices. The 5 major types of power-conversion devices are named, analyzed, evaluated, and discussed.

Part III: Applicability, Compatibility, and Potential Contributions to Tactical Vehicles. The areas covered in this part are indicated by the title.

Part IV: Tradeoff Analysis and Recommended Programs. The study findings are summarized and the results of a full evaluation of accumulated data are reviewed and presented.

Energy-Conversion Devices Considered

Energy-conversion devices were evaluated to find those that would offer technological advances appreciably improving the physical or performance characteristics of tactical vehicles in the foreseeable future. The engines studied are discussed in the following paragraphs, first those (the first nine) that are contenders for consideration or that offer specific advantages and then those judged unsuitable for use in tactical vehicles.

Spark-Ignition Reciprocating Engines. Spark-ignition engines will continue to be used in tactical vehicles in low power ranges since these engines are relatively compact and can be procured at a reasonable cost. Development of spark-ignition reciprocating engines for commercial applications will continue, but the Army will require continuing R&D funding to modify these engines for use in tactical vehicles.

Compression-Ignition Reciprocating Engines. Compression-ignition engines will continue to be developed by industry in power ranges to 600 hp for commercial application, but these engines must be modified for use in tactical vehicles. Industry has limited application for such lightweight engines in higher power ranges and therefore has no incentive to sponsor such development.

In order to have compression-ignition engines above 600 hp applicable to tactical vehicles, the US Army would have to sponsor their development. Little risk is involved in the development of compression-ignition engines at moderate specific weight and volume in power ranges above those now available.

Hybrid Engines. The hybrid engine represents a relatively new engine concept that combines the best features of the spark-ignition and compression-ignition reciprocating engines. The hybrid engine has the best potential for improving the capabilities of many future vehicles and when fully developed could be produced with the tooling and facilities now used by industry. However, full development of hybrid engines will require a lead time of from 5 to 10 years, depending on power level, and an element of risk is involved in this development.

Gas-Turbine Engines. The gas-turbine engine will offer excellent power-to-weight and power-to-size ratios for tactical vehicles when fully developed. Both the development and unit cost of gas turbines will be relatively high, but for vehicles that require an exceedingly high power output within weight and space limitations, such as a main battle tank, the cost is warranted. Gas-turbine engine development will require a relatively long lead time and holds an element of risk. However, when successfully developed, gas-turbine engines will replace compression-ignition engines for use in high power ranges.

Rotary-Piston Engines. The power output of rotary-piston engines is high when contrasted with their light weight and compactness. Their simple construction contributes to low unit production cost. This type of engine, more than any other, lends itself to a "throw-away" policy when major repairs or overhaul become necessary. A serious problem arises from the fact that the companies that are developing the most promising engines hold proprietary rights to them. The capability of industry to produce rotary-piston engines at a competitive cost will depend in large measure on the success of Government negotiations with these companies.

Dynastar Engines. The Dynastar engine, a reciprocating engine with peripheral opposed pistons, offers greater compactness and lighter weight than the more conventional reciprocating compression-ignition engines. Several prototypes have been successfully produced. The cost of developing the Dynastar will be the same as the cost of developing the compression-ignition reciprocating engine. The capability of industry to produce the Dynastar during full mobilization would be limited because production tooling is not available.

Compound Engines. Compound engines are basically compression-ignition reciprocating engines that incorporate a power turbine driven by the engine's exhaust gases. These engines have a lower fuel-consumption rate for power output than any other engines considered. However, compound-engine development and unit cost would be relatively high since the design of this engine is somewhat complex. If maximum fuel economy is of prime interest, development of the compound engine could be warranted. The capacity of industry to produce this engine during a period of full mobilization would be limited because of the number of gas-turbine-engine components used.

Free-Piston Engines. The free-piston engine, while offering excellent torque characteristics, low fuel consumption, and multifuel capabilities, is limited for use in tactical vehicles because of its high weight-to-output and size-to-output ratios. A relatively long lead time would be required to produce these engines in production quantities since considerable research is needed to yield an acceptable design. Further, production facilities and tooling are not available at present.

irling-Cycle Engines. The Stirling-cycle engine offers silent operation in lo ower ranges. The latest version of this engine incorporates a development called the "Dineen" process. A decrease in engine weight-to-output and size-to-output ratios is indicated, but there is doubt that the decrease will yield ratios more favorable than those of the more conventional engines. The unit production cost of the Stirling-cycle engine will be relatively high since the engine is complex. Further, the density for use of the Stirling engine in tactical vehicles is low. Only limited quantities of this engine could be produced during a period of full mobilization since production facilities and tooling are not available. However, the engine does offer silent operation, a feature not available in any other engine that has been found acceptable in other respects.

Ammonia-Fueled Engines. Because of the corrosive effect of ammonia fuel on many metals and its harmfulness to personnel, ammonia engines would be highly vulnerable in combat. Therefore the introduction of these engines into the military system would immediately impose two new logistics problems. First the total number of vehicles needed would increase since the ammonia-fueled vehicles could be used only under noncombat conditions. Other types of vehicles in the same horsepower class would be required for combat conditions. This requirement nullifies the advantage of a decreased hydrocarbon-fuel transportation burden. Second an entire new line of spare parts would be required, supply depots would have to accommodate for the issue of the new ammonia engines, and maintenance depots would have to gear for their repair. Further disadvantages of ammonia engines are the need for increased maintenance because of the danger of fumes and leaks, the greatly increased bulk and weight of ammonia fuel compared to gasoline, the need for pressurized fuel tanks, and the power loss (approximately 20 percent) that is presently experienced when hydrocarbon-fueled engines are converted to ammonia operation.

Fuel Cells. It has been determined that fuel cells will remain too bulky, heavy, and inefficient within the foreseeable future for use in tactical vehicles. Exploratory and advanced development of this device for mobility operation should be held in abeyance until successful application is attained in stationary equipment. If fuel cells are successfully used in the future in stationary equipment and if many of the present problems are resolved, R&D of these cells for tactical vehicle applications should be reconsidered.

Unique Energy-Conversion Devices. Nine unique energy-conversion devices that were considered in the course of this study were determined to be much too heavy, excessively bulky, and too inefficient for application in tactical vehicles. Development of unique devices within the foreseeable future would not result in an energy-conversion device that would improve the capability of tactical vehicles.

Nuclear Reactors. Nuclear reactors will remain too bulky and too heavy for application in tactical vehicles. Further, materials to provide efficient shielding must be developed and a concept devised to overcome personnel and vehicle vulnerability to radiation hazards. In addition the cost of such devices must be substantially reduced.

Batteries. Batteries have application in tactical vehicles where silent operation is mandatory but a limited operating range is acceptable. For most tactical-vehicle operations, however, the operating range provided by batteries is too limited, and batteries are too bulky to be considered for such vehicles.

KGG Cycle or Kuhns Gasifier. Currently the Kuhns gasifier turbine engine is only a concept and its application to tactical vehicles lies too far in the future for consideration at this time. The successful development of this device should first be achieved for use in aircraft. At that time each feasible additional development should be considered for tactical-vehicle application.

Steam Engine. To date a concept has not been devised that would reduce the weight and size of a steam engine below that of present-day conventional reciprocating engines. Further consideration should not be given the steam engine unless the development of a vehicle offering silent operation is considered more important than overall size and weight limitations, and high torque characteristics are required at low power output.

Power-Conversion Devices Considered

Mechanical Power-Conversion Devices. Mechanical power-conversion devices have been used with success for many years in tactical as well as commercial wheeled and tracked vehicles. Today most mechanical power-conversion devices have outlived their usefulness for tactical-vehicle application. An exception is the synchromesh device, which will continue to have military application in low-powered vehicles with less than 120 hp. Industry is continually improving synchromesh power-conversion devices in various horsepower ranges. The Government could therefore make use of those commercially available units that fall within the applicable military power ranges by modifying these units to meet specific tactical-vehicle requirements.

The belt drive is another mechanical power-conversion device that has been investigated. This device can be produced at a low unit cost and would provide ease of operation at relatively slow speeds. Belt-drive units are used in commercial vehicles but have not as yet been proved acceptable for use in tactical vehicles. However, they appear to have potential for successful application in some types of small tactical vehicles with power ranges up to 50 hp.

Hydrokinetic Transmissions and Power Trains. Hydramatic* and Torqmatic* power-conversion devices are widely used for commercial application. They have also been employed in the past for installation in certain tactical vehicles. However, only the Torqmatic devices have remained in the system, and their use is also restricted to relatively few kinds of tactical vehicles. Torque-converter planetary-gear power-conversion devices are successfully used in both tactical wheeled and tracked vehicles. They provide vehicles with performance capabilities superior to those achieved with other hydrokinetic

*"Hydramatic" and "Torqmatic" are trade names of General Motors Corporation (GMC) transmissions.

devices of earlier design. They are available in all required power ranges for tactical wheeled vehicles but cover only power ranges up to 600 hp for tactical tracked vehicles. Therefore R&D of these devices is required to extend their range and to furnish some improvement to existing units. Full development of hydrokinetic power trains will assure coverage of all required power classes for tactical tracked vehicles until more advanced devices are developed and proved acceptable.

Hydromechanical Power-Conversion Devices. Hydromechanical power-conversion devices are relatively new in concept, but the few prototypes that have been developed promise to exceed the capabilities of present hydrokinetic power-conversion devices.

Since a hydromechanical power train is basically made up of two symmetrically opposite hydromechanical transmissions, development of the power trains will benefit transmission development. Hydromechanical power-conversion devices would offer improved specific-weight and volume-to-torque ratios, better acceleration, and greater fuel economy. Their successful development would result in a unit that could be readily applied in various systems with components that would be common to both wheeled and tracked vehicles. Thus the logistics burden would be reduced. However, a long lead time will be required, and there are some elements of risk for successful development of these devices in all power ranges. When fully developed, hydromechanical power-conversion devices will replace most of the hydrokinetic units, which will remain in the system during the transition period.

Hydrostatic Power-Conversion Devices. Hydrostatic power-conversion devices are being developed by industry in the narrow speed-torque range for slow-moving commercial vehicles. Devices operating in this range are also applicable to special-purpose tactical vehicles that do not readily permit the installation of more conventional devices because of space limitations or otherwise cumbersome mechanical-drive-line component arrangements. Hydrostatic power-conversion devices in the medium speed-torque range find only few commercial applications.

Industry's interest in pursuing these developments is therefore limited and does not extend beyond that of keeping abreast with the state of the art. Both narrow- and medium-range hydrostatic devices have been installed in several types of tactical vehicles, but with only marginal success. However, a need does exist for this type of power-conversion device for larger, faster-moving military vehicles requiring additional functions from the power-conversion device that cannot be accomplished efficiently with strictly mechanical components, such as delivery of power to the wheels of amphibious vehicles for steering and wheel retraction.

Electric-Drive Power-Conversion Devices. The inherent features and capabilities of both dc and ac electric-drive systems would improve the physical and performance characteristics of some types of tactical vehicles. Electric-drive power-conversion devices have been used with success in a

TABLE S-1

Energy- and Power-Conversion Devices and Their Potential Contributions to Tactical Vehicles

Type of device	Contribution to physical improvement of vehicle	Contribution to operational improvement of vehicle
Energy-Conversion Devices		
Rotary spark-ignition engine	Reduces vehicle weight; provides more space for cargo and personnel	Decreases operational cost; decreases vehicle maintenance requirements; reduces logistic requirements
Dynastar compression-ignition engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Variable-compression-ratio (VCR) engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Very-high-output(VHO) engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Extremely-high-output (EHO) engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Hybrid engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy; contributes prompt mobilization capacity
Stirling-cycle engine	—	Offers silent operation
Gas-turbine engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Piston-turbine compound engine	Reduces vehicle weight; offers more space for cargo and personnel	Improves mobility; decreases operational cost; complies with fuel policy
Free-piston turbine engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Power-Conversion Devices		
Torque-converter planetary gear, TX series	Reduces vehicle weight	Improves mobility; improves controllability; decreases logistic requirements; decreases vehicle maintenance requirements; increases vehicle reliability; decreases operational cost
Torque-converter planetary gear, X series	Reduces vehicle weight; offers more space for cargo and personnel	Improves mobility; improves controllability; decreases logistic requirements; decreases vehicle maintenance requirements; increases vehicle reliability; decreases operational cost
Belt drive	Provides better load distribution on axles; increases vehicle ground clearance; readily adaptable to various small engines	Decreases logistic requirements

TABLE. S-1 (continued)

Type of device	Contribution to physicol improvement of vehicle	Contribution to operotionol improvement of vehicle
Hydrostatic drive	Offers flexibility of design; provides better load distribution on axles and increased ground clearance for wheeled vehicles; provides more space for cargo and personnel; readily adaptable to various engines	Improves mobility and ease of operation; improves controllability; reduces vehicle maintenance requirements; decreases logistic requirements; reduces operational cost
Hydromechanical drive	Reduces vehicle weight; offers more space for cargo and personnel; readily adaptable to various engines	Improves mobility and ease of operation; improves controllability; reduces vehicle maintenance requirements; decreases logistic requirements; reduces operational cost
Electric drive	Offers flexibility of design; provides better axle-loading ratios and increased ground clearance; provides more space for cargo and personnel; readily adaptable to various engines	Improves mobility and ease of operation; provides better vehicle control; reduces vehicle maintenance; decreases logistic requirements; reduces operational cost

number of special commercial vehicles but have yet to achieve success in tactical-vehicle application. There is a need for the electric-drive devices in special types of tactical venicles, but their development must continue to reduce the unit size, weight, and cost. This study has revealed that so far greater success has been experienced with the development of dc electric-drive systems than with ac electric-drive systems. Direct-current electric-drive systems are being developed by industry for commercial vehicles not required to operate under environmental conditions as severe as those to which tactical vehicles are exposed. Available commercial dc electric-drive systems should be modified for application to special tactical vehicles whose configuration does not permit the ready use of more conventional power-conversion devices. Acceptable ac electric-drive systems have not been developed by industry for commercial vehicle applications. Alternating-current electric-drive systems have the potential of achieving a weight and size reduction greater than that of comparable dc electric-drive systems. They would have greater vehicular applications than the dc system but should be installed in vehicles specifically designed to take full advantage of ac systems. Based on the findings of this study the most opportune course to pursue would be to develop dc electric-drive systems first but at the same time to continue R&D of ac electric-drive systems. The dc electric-drive system could be available for tactical vehicles within the 1970–1975 time frame, and development of ac electric-drive systems could be accomplished within the 1975–1985 time frame.

A summary of the potential contributions of energy- and power-conversion devices to the improvement of tactical vehicles is given in Table S-1.

SUMMARY

Conclusions

Energy-Conversion Devices—First Priority

1. Spark-Ignition Engines: US Army modification of commercially available engines in power ranges to 250 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	2,500,000	900	315,350
120–250	2,500,000	1,400	241,100

Lead time required for full production: 4 years
Probability of successful development: Excellent
Mobilization capacity: Excellent
Remarks: Relatively compact, reasonable cost, proved reliability.

2. Compression-Ignition Engines: US Army development of commercially available engines in power ranges under 500 hp and of new engines in power ranges over 500 hp consolidated into a new R&D program is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
250–500	5,000,000	8,000	77,900
500–1000	12,000,000	19,000	34,450
≥1000	17,000,000	24,000	17,000

Lead time required for full production: 7 years
Probability of successful development: Excellent
Mobilization capacity: Good
Remarks: Reliable, lower fuel-consumption rate than spark-ignition engine, multifuel capability.

3. Hybrid Engines: US Army development in power ranges 100 to 1000 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	7,000,000	2,100	315,350
120–250	9,000,000	4,000	241,100
250–500	14,000,000	8,000	77,900
500–1000	17,000,000	16,000	34,450

Lead time required for full production: 10 years
Probability of successful development: Good
Mobilization capacity: Excellent
Remarks: Combines best features of spark-ignition and compression-ignition reciprocating engines; good specific fuel consumption; multifuel capabilities.

4. Gas-Turbine Engines: US Army development in power ranges over 1000 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
500–1000	17,000,000	19,000	34,450
≥1000	25,000,000	23,000	17,000

Lead time required for full production: 12 years
Probability of successful development: Good
Mobilization capacity: Fair
Remarks: Excellent power-to-weight and power-to-size ratios, excellent torque characteristics, multifuel capabilities.

5. The preceding four engines (listed in order of priority) promise greater across-the-board benefit from R&D than all the other engines listed or considered.

6. These four engines represent the minimum number that should be developed.

Energy-Conversion Devices—Second Priority

7. Rotary-Piston Engines: US Army development in power ranges to 250 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	2,000,000	450	315,350
120–250	3,500,000	750	241,100

Lead time required for full production: 9 years
Probability of successful development: Good
Mobilization capacity: Good
Remarks: Low unit cost, best potential for throw-away concept, good maintainability, multifuel capabilities.

8. Dynastar Engines: US Army development in power ranges from 120 to 250 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
120–250	4,000,000	5,000	241,100
250–500	6,000,000	10,500	77,900
500–1000	11,000,000	23,000	34,450

Lead time required for full production: 9 years
Probability of successful development: Good
Mobilization capacity: Good
Remarks: Highly compact engine, excellent family capabilities.

9. Compound Engines: US Army development in power ranges from 250 to 1000 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
120–250	12,000,000	6,500	241,000
250–500	16,000,000	10,000	77,900
500–1000	18,000,000	20,000	34,450

Lead time required for full production: 9 years
Probability of successful development: Good
Mobilization capacity: Fair
Remarks: Best engine for maximum fuel economy, multifuel capabilities.

10. Free-Piston Engines: US Army development in power ranges from 120 to 1000 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
120–250	12,000,000	6,500	241,100
250–500	16,000,000	10,000	77,900
500–1000	18,000,000	20,000	34,450

Lead time required for full production: 13 years
Probability of successful development: Good
Mobilization capacity: Fair
Remarks: Excellent torque characteristics, low fuel consumption, multifuel capabilities.

11. Stirling-Cycle Engines: US Army development in power ranges to 120 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	6,000,000	1,400	88,600

Lead time required for full production: 9 years
Probability of successful development: Poor
Mobilization capacity: Poor
Remarks: Offers silent operation.

12. Engine technology will benefit less from R&D on the preceding five engines than it will from R&D on the first four.

13. The following energy-conversion devices are not applicable for use in tactical vehicles:

(a) Ammonia-fueled engines
(b) Fuel cells
(c) Unique energy-conversion devices
(d) Nuclear reactors
(e) Batteries
(f) Kuhns gasifier
(g) Steam engines

Power-Conversion Devices*

14. Torque-Converter Planetary-Gear Power Trains: US Army development in power ranges above 600 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
500–1000	7,000,000	9,500	34,100
>1000	9,000,000	14,500	17,000

Lead time required for full production: 6 years
Probability of successful development: Excellent
Mobilization capacity: Good
Remarks: Compact, ease of operation, good family capabilities.

15. Belt-Drive Transmissions: US Army development in power ranges to 120 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
<120	1,750,000	400	94,600

Lead time required for full production: 4 years
Probability of successful development: Good
Mobilization capacity: Excellent
Remarks: Low unit cost, automatic speed-torque change, readily maintained.

*Not in order of priority.

16. Hydromechanical Power Trains: US Army development in all power ranges is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	2,500,000	4,000	24,500
120–250	3,500,000	4,500	34,500
250–500	4,500,000	5,000	76,900
500–1000	6,000,000	6,500	34,100
≥1000	9,500,000	8,200	17,000

Lead time required for full production: 10 years
Probability of successful development: Good
Mobilization capacity: Good
Remarks: Permits engine to operate at its most economical power range; continuous variable-controlled vehicle speed; lower unit costs than comparable power trains; development of power trains would be applicable to transmissions.

17. Hydrostatic (Narrow-Range) Systems: US Army development in power ranges under 250 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	640,000	2,200	94,600
120–250	680,000	3,500	2,500

Lead time required for full production: 6 years
Probability of successful development: Good
Mobilization capacity: Good
Remarks: Design flexibility; permits engine to operate at its most economical power range; continuous variable-controlled vehicle speed; ease of control.

18. Hydrostatic (Medium-Range) Systems: US Army development in power ranges to 1000 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	1,000,000	9,800	184,650
120–250	2,200,000	11,100	68,400
250–500	2,320,000	14,760	1,000
500–1000	2,920,000	18,700	350
≥1000	3,430,000	15,800	17,000

Lead time required for full production: 12 years
Probability of successful development: Poor
Mobilization capacity: Fair
Remarks: Design flexibility; permits engine to run at its most economical power range; continuous variable-controlled vehicle speed; ease of control.

19. Electric-Drive (Direct-Current) Systems: US Army development in power ranges to 250 hp is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	1,250,000	4,800	141,600
120–250	2,000,000	8,600	64,000

Lead time required for full production: 8 years
Probability of successful development: Good
Mobilization capacity: Poor
Remarks: Design flexibility; permits engine to operate at its most economical power range; continuous variable-controlled vehicle speed; ease of operation.

20. Electric-Drive (Alternating-Current) Systems: US Army development in all power ranges is warranted.

Horsepower class	Development costs, dollars	Estimated unit costs, dollars	Density
≤120	3,700,000	8,000	90,050
120–250	3,900,000	15,000	66,000
250–500	4,200,000	35,000	1,000
500–1000	4,400,000	50,000	850
≥1000	5,600,000	60,500	17,000

Lead time required for full production: 14 years
Probability of successful development: Good
Mobilization capacity: Poor
Remarks: Design flexibility; permits engine to operate at its most economical power range; continuous variable-controlled vehicle speed; provides auxiliary power source; ease of operation.

Availability Times

21. The four energy-conversion devices for which the estimated availability times are shown in Fig. S-1 make the greatest across-the-board contribution toward satisfying the requirements of most military tactical vehicles in all power ranges. The five energy-conversion devices in Fig. S-2 each have a specific and predominant feature that is not offered by the four treated in Fig. S-1 but that may be required to meet specific requirements of vehicles of a special type. Estimated availability times shown in Figs. S-1 and S-2 are based on the assumption that full development effort is continued or initiated by the next fiscal year after publication of this report. Any additional elapsed time before initiation of development would most likely be additive to the estimated time frames.

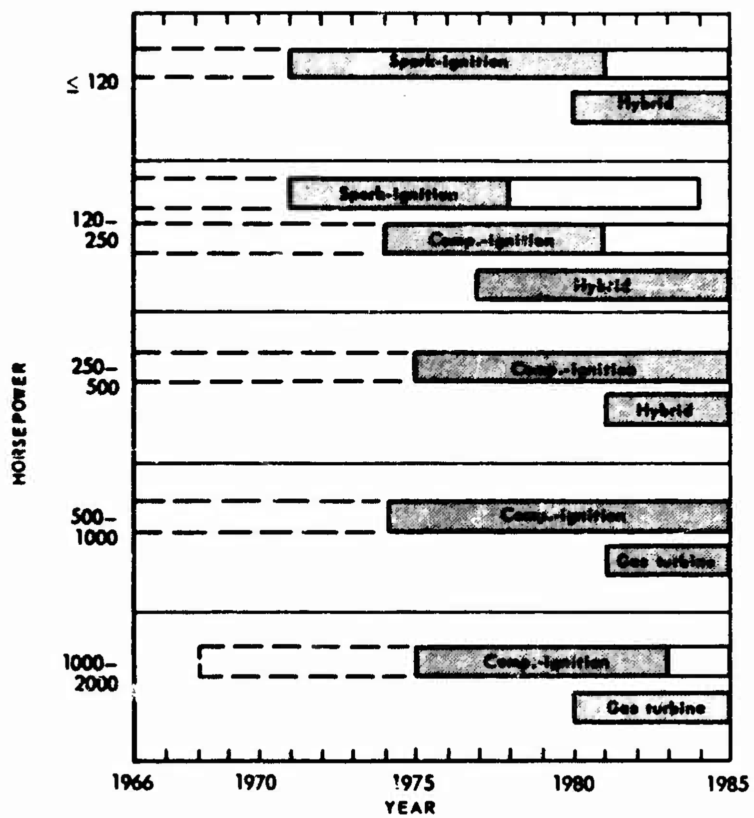


Fig. S-1—Energy-Conversion Devices for Most Tactical Vehicles

Available without future R&D
Available with new R&D
Available if required

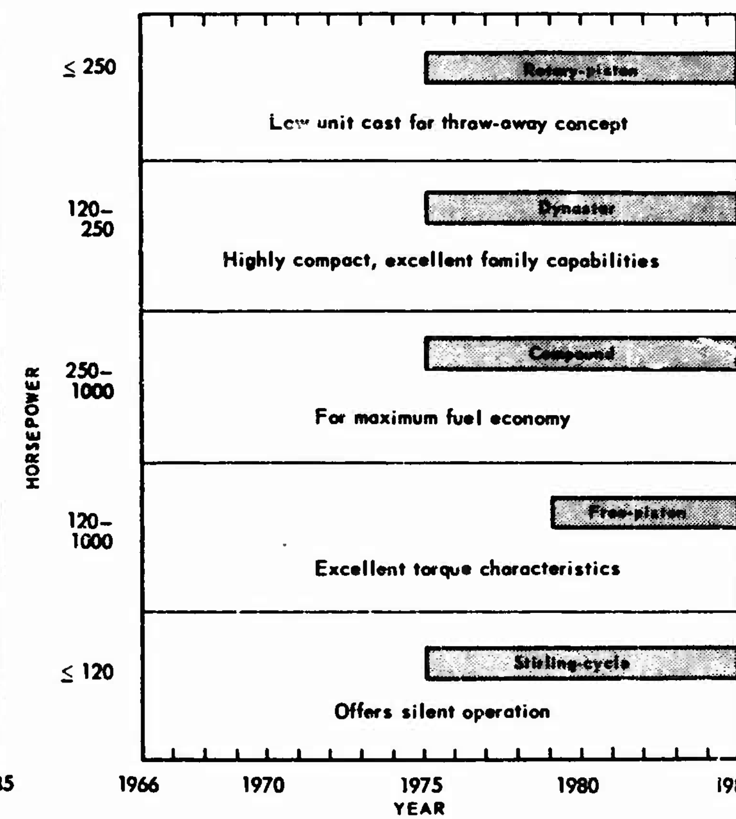


Fig. S-2—Energy-Conversion Devices for Special Vehicle Requirements

Available without future R&D
Available with new R&D
Available if required

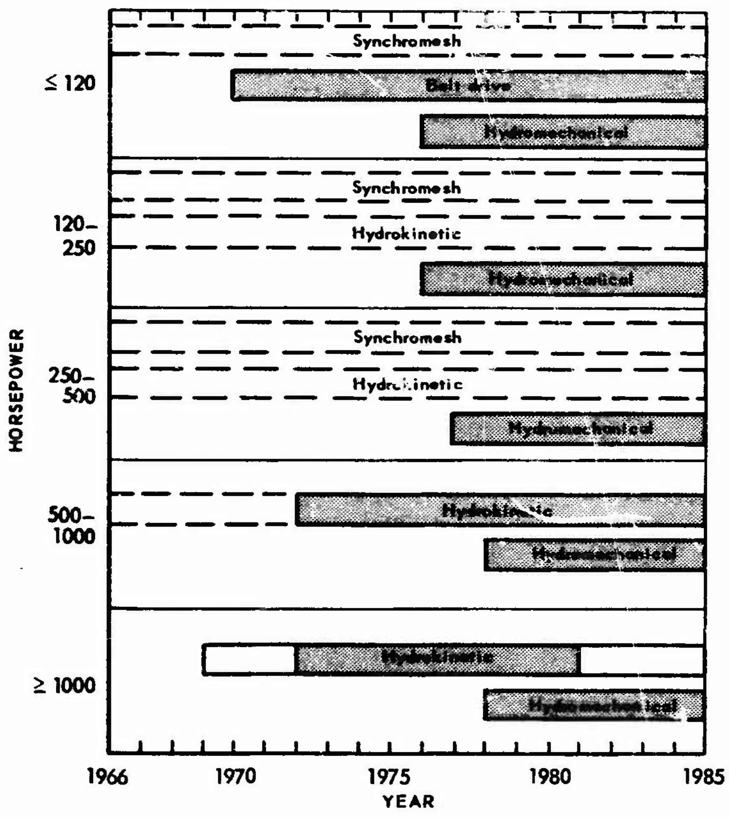


Fig. S-3—Power-Conversion Devices for Most Tactical Vehicles

- Available without future R&D
- Avoilable with new R&D
- Avoilable if required

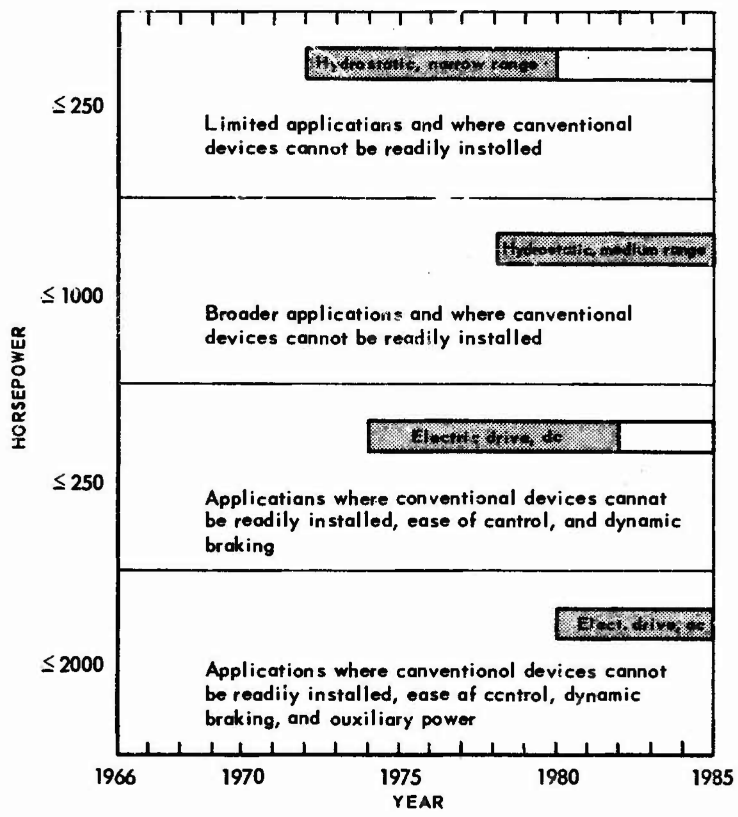


Fig. S-4—Power-Conversion Devices for Special Vehicle Requirements

- Available without future R&D
- Available with new R&D
- Available if required

22. The mechanical, hydrokinetic, and hydromechanical power-conversion devices for which the estimated availability times are shown in Fig. S-3 make the greatest contribution toward satisfying the requirements of most military tactical vehicles in all power ranges. The electric and hydrostatic drive systems treated in Fig. S-4 offer design flexibility where conventional devices cannot be readily installed without considerable vehicle design compromise. Estimated availability times shown in Figs. S-3 and S-4 are based on the assumption that full development effort is continued or initiated by the next fiscal year after publication of this report. Any additional elapsed time before initiation of development would most likely be additive to the estimated time frames.

Engine-Transmission Power Plants for Tactical Vehicles

ABBREVIATIONS

ac	alternating current
ATAC	Army Tank-Automotive Center
bdc	bottom-dead-center
BEST	Ballastable Sectionalized Tractor
bhp	brake horsepower (only with arabic numbers)
BICERA	British Internal Combustion Engine Research Association
bmep	brake mean effective pressure
bsfc	brake specific fuel consumption
CAE	Continental Aviation and Engineering Corporation
CIE	compression-ignition-engine
CITE	compression-ignition-turbine-engine (fuel)
dc	direct current
EHO	extremely high output
FRG	Federal Republic of Germany
ghp	gasoline horsepower (only with numbers)
GMC	General Motors Corporation
GVW	gross vehicle weight
ihp	indicated horsepower
imep	indicated mean effective pressure
KGG	cycle or "KHUNS" Gasifier
MAN	hypercycle combustion principle
MHD	magnetohydrodynamic
OTAC	Ordnance Tank-Automotive Command (now ATAC)
PTC	piston-turbine compound engine
R&D	research and development
SIGMA	acronym from name of French manufacturing company
S.I.	spark-ignition reciprocating engines
SSFC	shaft specific fuel consumption
TCP	Texaco combustion process
UET	Universal Engineering Tractor
USAERDL	United States Army Engineering Research and Development Laboratory
USATAC	United States Army Tank-Automotive Center
VCR	variable compression ratio
VHO	very high output
WSP	Witzky swirl-stratify combustion process
WWII	World War II

Chapter 1

INTRODUCTION TO THE STUDY

The purpose of this study is to identify those areas that show promise of yielding the greatest potential return from investment in R&D of energy- and power-conversion devices for use in future tactical vehicles. The Army desires to develop conversion devices and supporting components that will not be developed commercially owing to their restricted military use. This study seeks to establish a basis from which priorities can be determined and, since funds are limited, duplication of expense and effort can be avoided.

Data on present energy- and power-conversion devices and related systems were obtained from Government agencies, industry, universities, and nonprofit organizations to supplement the authors' knowledge of design and application of components for military vehicles. These data were analyzed to assess the status of present technology and R&D programs as well as that of new concepts or anticipated programs.

Although analysis and discussion of the components and systems may encompass a broad time frame, their availability is scheduled for use in tactical vehicles fielded in the 1975–1980 and 1980–1985 time frames. The vehicles—wheeled, tracked, special-purpose, and amphibious—are grouped into specific classes by horsepower and weight as shown in the accompanying tabulation.

Horsepower	Vehicle weight, tons
≤120	≤5
120–250	5–15
250–500	15–35
500–1000	35–50
≥1000	≥50

Although some mention is made of conversion devices suitable for use in aircraft and stationary equipment, these are not considered in detail in this study.

The study is based on the premise that superior tactical vehicles can be developed through technological advancement of energy- and power-conversion devices. Ten criteria considered in determining technological advancement are:

(a) Reduction in size
(b) Reduction in weight

(c) Increased performance
(d) Increased reliability and decreased maintenance
(e) Reduced initial cost
(f) Reduced operating cost
(g) Compatibility with US Army fuel policy
(h) Flexibility of vehicle design
(i) Industry's mobilization capacity
(j) Reduction of proprietary components

This report is divided into four parts. Part I discusses energy-conversion devices and Part II treats of power-conversion devices. The compatibility of specific energy-conversion devices with specific power-conversion devices is considered in Part III. Part IV discusses the use of various systems in specific classes of tactical vehicles. From analysis of findings and consideration of various tradeoffs, conclusions are drawn and recommendations are presented.

PART I

Energy-Conversion Devices

Chapter 2

INTRODUCTION TO PART I

An energy-conversion device converts stored or potential energy into mechanical, thermal, chemical, or electrical energy for useful work. For practical application in vehicles these devices must have physical characteristics that can be accepted by the vehicle and be able to produce power at a rate that can be controlled by the operator. In this study these devices include engines, batteries, fuel cells, reactors, and unique energy-conversion devices. Steadily increasing vehicle performance requirements have continued to place higher demands on energy-conversion devices, which has resulted in the development of new or improved devices.

From 1943 to 1965 significant improvements were made in tactical vehicles. Some of the more significant improvements are listed in the accompanying tabulation.

Improvement area	Change, %
Fuel economy, miles/gal	+180
Hours per battlefield day	+350
Cruising range, miles	+250
Power-to-weight ratio, hp/ton GVW[a]	+225
Vehicle weight, lb	− 35
Reliability and durability	+ 50

[a]Gross vehicle weight.

To obtain superior tactical vehicles the military are continually upgrading the physical and performance requirements for new vehicles. Many types of tactical wheeled vehicles must now be capable of swimming in inland waters. Some amphibians have water speeds two or three times those of their predecessors. Most new tactical vehicles are lighter and exert lower ground pressures (which increases their mobility) and are more easily transported by air. They are also capable of greater speeds and faster acceleration, which improve their agility. Figures I-1 to I-4 illustrate the steadily increasing power-to-weight ratio of several classes of tactical vehicles since WWII. To meet increased vehicle operating-range requirements and probable scarcity of some fuels during wartime, greater fuel economy and multifuel operation are required. These improvements have affected both initial and operating costs. Such requirements

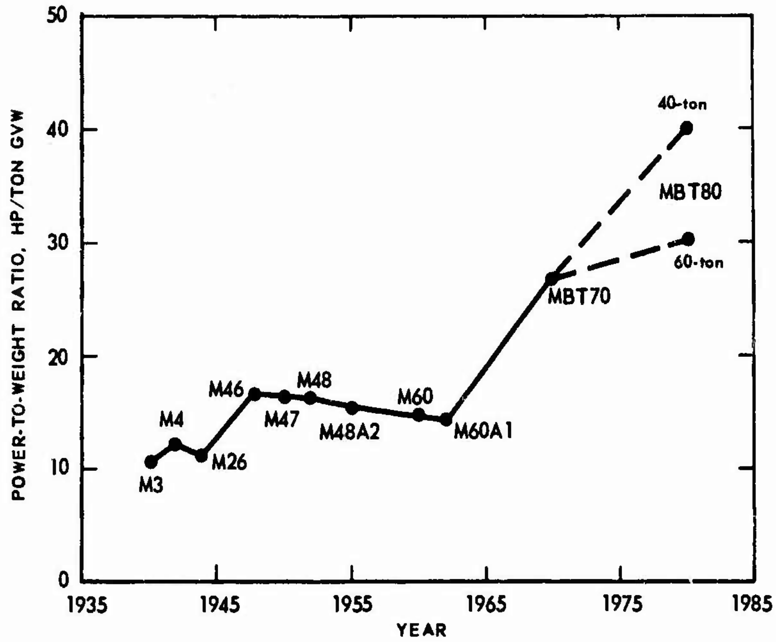


Fig. I-1—Evolution of Power Requirements for Battle Tanks

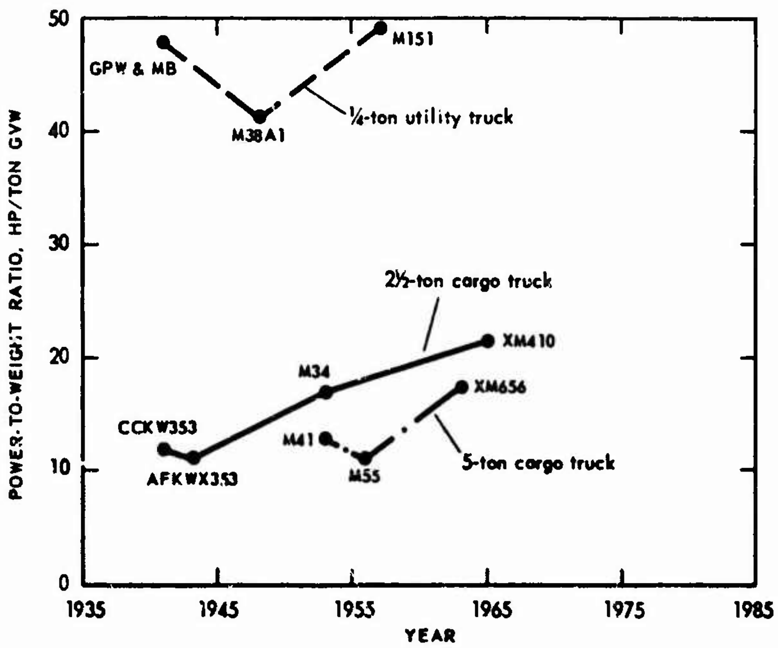


Fig. I-2—Evolution of Power Requirements for Cargo Trucks

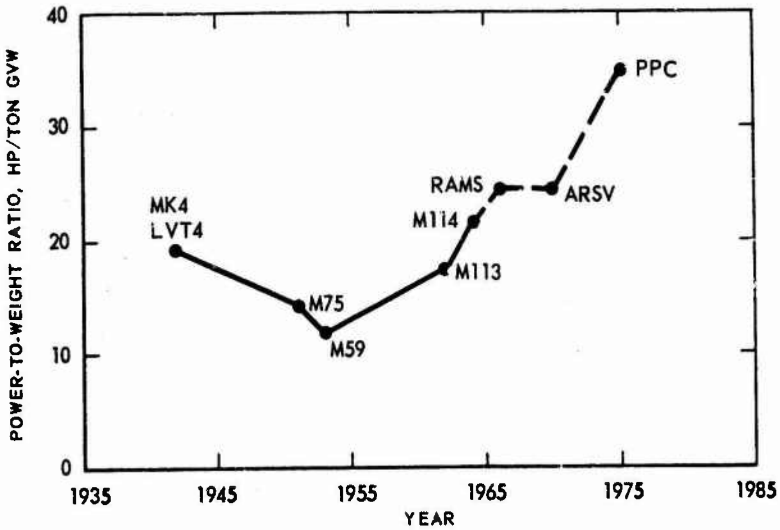


Fig. I-3—Evolution of Power Requirements for Armored Personnel Carriers

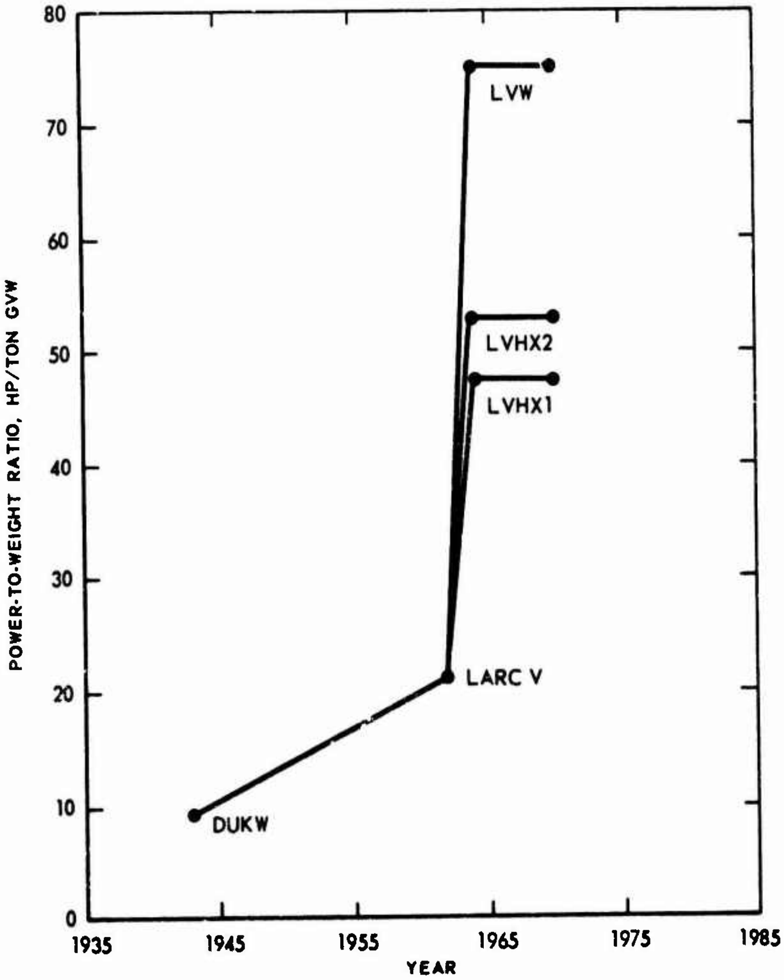


Fig. I-4—Evolution of Power Requirements for Amphibious Lighters could not have been met without the development of components that thus became available to the vehicle designer.

To meet these requirements a large number of energy-conversion devices must be analyzed to determine which ones have the greatest potential for development.

Vehicle improvements cannot be accomplished without the continued development of energy-conversion devices well in advance of the development of vehicles. Past experience has shown that on the average 5 years are required to develop a new energy-conversion device and that an additional 5 years are required to develop a new vehicle. Therefore no new conceptual device can be fielded prior to 10 years from the time of its initial development, since to develop a vehicle without assurance that the device intended for it will prove acceptable would involve too great a risk. Predictions for the development of energy-conversion devices in this report are for the time frame in which they will be available for the designer to consider for use when a vehicle-development program is initiated.

Chapter 3

RECIPROCATING-PISTON ENGINES

The reciprocating-piston engine has been used exclusively for many years as a power source for tactical-vehicle propulsion. Until the last decade the spark-ignition gasoline engine has been used almost exclusively for tactical vehicles. These engines were modified passenger-car engines (sometimes used in multiples of two to five units in tanks and tracked vehicles) or adapted or modified aircraft engines. However, the fuel consumption of spark-ignition gasoline engines is high, particularly during idling and low-load conditions, which make up a large part of the operating time of military vehicles in combat areas. Because the diesel engine demonstrated greater fuel economy than the gasoline engine, at the onset of WWII the Armored Force Board decided to pursue development of the diesel engine for use in tanks to ease the fuel-supply burden.

Several diesel engines developed by industry were considered. Two commercial six-cylinder in-line engines connected to a common gearbox were used in the M4A2 tank. The others were radial-type engines, of which one was a dieselized version of an aircraft engine, and the other was a new design based on what was known about radial aircraft engines. The former was in limited production. However, just prior to production of the radial engines, the Assistant Chief of Staff, G4, and the Armored Force Board in February 1942 reversed their position on the utilization of diesel engines and ruled that gasoline fuel would be used universally. The reason for this reversal was that, although adequate supplies of diesel fuel existed in the US, there was an insufficient amount of this fuel in the theaters of war. Another major factor that influenced their decision was that maintenance and supply problems would be greatly increased where both diesel fuel and gasoline were required for combat vehicles. As a result of this decision the use and development of diesel engines were minimal.

During the years between WWII and the Korean War, militarized heavy-duty commercial spark-ignition gasoline engines were used in wheeled vehicles. A new series of air-cooled spark-ignition gasoline engines, whose development was initiated in WWII (1943), was incorporated into tanks and other tracked and special-purpose vehicles. These new engines were based on several cylinder-bore sizes to offer "family" capabilities with complete power coverage to 1000 hp and with maximum interchangeability of parts and components. These new engines were lightweight and compact and were produced in both air-cooled opposed (AO series) and air-cooled V (AV series) configurations.

During the Korean War, combat operations indicated that increased vehicle cruising range was desired. The achievement of range increase dictated that greater fuel economy must be obtained from vehicle power plants. Realizing the diesel engine's superior fuel economy and potential multifuel capabilities and the fact that very little could be done to improve the spark-ignition gasoline engine, the Ordnance Corps investigated the compression-ignition engine. This investigation resulted in a decision to adopt the compression-ignition engine as the main power source that would replace all spark-ignition gasoline engines in new vehicles requiring 100 to 175 hp (depending on the type of vehicle) or more. The Ordnance Corps solicited proposals from the engine industries in 1951 for a suitable compression-ignition tank engine, and contracts were awarded to two companies in 1953–1954 to develop an X-configuration and an H-configuration compression-ignition engine. However, these programs were terminated owing to the probability of greater success in attempting to convert an Ordnance Corps AV-1790 series, 12-cylinder 90-deg-V 4-stroke-cycle air-cooled spark-ignition engine to the diesel-cycle. This program, initiated in 1954, resulted in the successful development and production of the AVDS-1790-2 engine for the M60 105-mm-gun main battle tank.

In making the decision to adapt the compression-ignition engine for all but the smallest high-density vehicles, the military also realized that a shortage in the supply of high-cetane middle-distillate diesel oils during any future war was probable. This is due to the large increase in the use of diesel oil in the past few years by aircraft, naval vessels, merchant ships, small craft, locomotives, tractors, and highway trucks. Because of this the Department of the Army required all compression-ignition engines to be able to operate on CITE (compression-ignition-turbine-engine) fuel, which is a JP-4-type (jet-propulsion) fuel, in addition to diesel fuel, with the ultimate goal that all compression-ignition engines would have multifuel capabilities. The reasoning is that the JP-4-type fuel, with a potential of 40 to 50 percent of the yield from a barrel of average crude petroleum, is more widely available than diesel fuel. The range of properties of JP-4-type fuel is wide. The referee-grade fuel (MIL-F-45121) selected by the Army for compression-ignition engines is described as a hydrocarbon fuel oil having 40 to 60 percent (by volume) catalytic-cracked components and the remainder straight-run petroleum components, with a distillation end point between that of gasoline and diesel fuel and a maximum cetane number of 35 (in contrast to the cetane rating of approximately 40 to 45 for diesel fuel). It is also characterized by low vapor pressure to reduce evaporation losses and the likelihood of vapor lock. Its potential availability is much higher than that of diesel fuel since it constitutes 40 to 50 percent of the average barrel of crude petroleum. The relation between octane ratings and cetane ratings is illustrated in Fig. I-5.

Four types of hydrocarbon fuels can be expected to be generally available for use in tactical ground vehicles: 86/95 (supersedes 83/91) octane combat gasoline, diesel-grade DF-A fuel, CITE fuel, and JP-4 fuel. Fuel logistic burdens could be reduced considerably if military compression-ignition engines had multifuel capabilities and could operate on a wide range of fuels from combat gasoline to the heavy fuel oils. However, the problems in developing a multifuel engine arise from the difficulty of obtaining satisfactory performance when operating on military-grade combat gasoline, which has an octane rating of 86/95 (86 by the motor method and 95 by the research method) and a very low cetane rating of approximately 28. Because of the low cetane rating of this fuel, it is very difficult to ignite by diesel-cycle compression heat. The problems are generally uncontrolled combustion, a faster rise in pressure, and poorer conversion of the energy in the fuel into work than with a fuel of higher cetane rating.

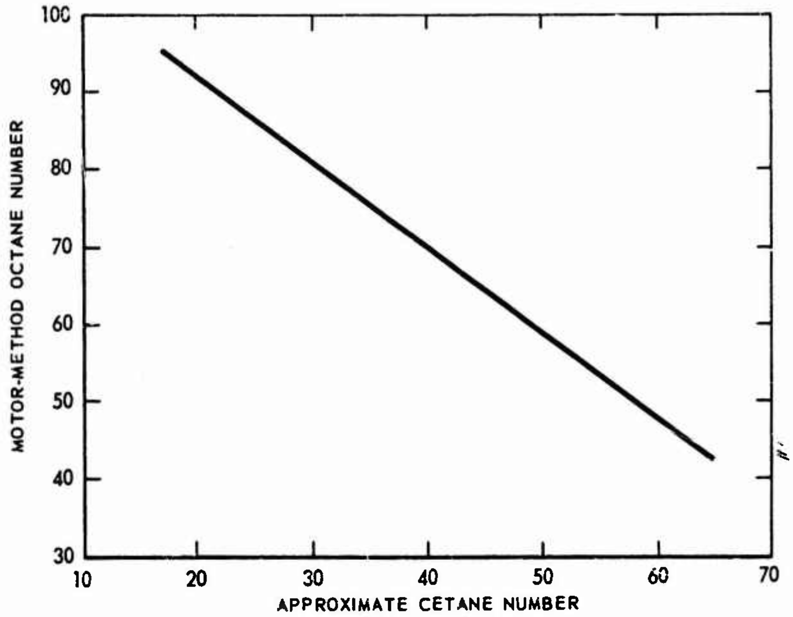


Fig. I-5—Relation between Octane and Cetane Ratings

50 percent blends of diesel and isooctane fuels.

The US Army, pursuing the goal of achieving a multifuel capability in compression-ignition engines, initiated a program in 1955 to develop a multifuel engine based on the MAN, or "hypercycle," combustion system. The successful development of this multifuel engine resulted in production and incorporation of a naturally aspirated version in the 2½-ton cargo truck, and a turbocharged version of the same basic engine in the 5-ton cargo truck. In the late fifties and early sixties the US Army pursued the development of several advanced engine-combustion concepts in keeping with its new policy to attain multifuel capabilities in reciprocating piston engines above 100 hp for future tactical vehicles. Although each of these engine concepts contains certain unique features, the ultimate technological goals of all are the same, i.e., high power output, low weight, compactness, low fuel consumption, high reliability, and durability—all at a minimum cost. Present development projects include the following engines:

(a) Variable compression ratio (VCR)
(b) Very high output (VHO)

(c) Extremely high output (EHO)
(d) AVM series
(e) Hybrid

Until the successful development and availability of one or more of these advanced engines the military is utilizing proved heavy-duty commercial compression-ignition engines and spark-ignition gasoline engines (in small vehicles up to 160 hp), in addition to the military-designed engines, in tactical-vehicle applications.

The reciprocating-piston compression-ignition engine, more commonly known as the "diesel" engine, has become the primary source of power for tactical vehicles. The principal reasons for this choice are its favorable performance characteristics, reliability and durability, and economy of operation. Also of major importance is its multifuel potential. The majority of leading compression-ignition engines today have at least a limited multifuel capability and a potential for development of full multifuel capabilities, including the efficient burning of combat gasoline. One series of military-designed engine, the LD-465 and LDS-465, which is currently in production for use in the 2½- and 5-ton cargo trucks, has a complete multifuel capability. In its simplest interpretation the term "multifuel" designates an engine with the capability of burning more than one fuel. In a sense, then, most of the leading compression-ignition engines today are multifuel engines because they burn a range of fuels from No. 2 diesel to Army compression-ignition-engine fuel (CIE) without modification. However, it is considered that the military desire in regard to multifuel engines is that the engine be capable of burning No. 2 diesel-grade DF-A fuel (MIL-VV-F-800), CITE (MIL-F-46005), grade JP-4 jet fuel (MIL-J-5624), and 86/95 combat gasoline (MIL-G-3056). It is also desired that the engines be capable of burning any and all mixtures of these and other intermediate fuels such as kerosene, No. 1 diesel, and the JP series without engine modification or adjustment.

Development work on compression-ignition engines has shown that the problems of burning and handling fuels increase as the specific gravity of the fuels decreases. The real problem has been one of extending the range of fuels the engine will handle. The ability of the engine to burn a wide range of fuels is primarily a matter of (a) handling the lighter fuels and controlling their introduction into the engine combustion chamber, and (b) obtaining full performance (equal to diesel fuel) with combat gasoline when both types of fuel are used. This problem can be solved through the incorporation of properly designed fuel-injection systems, increased compression ratio, direct-injection toroidal combustion chambers in the piston crown, and the use of an automatic fuel-density compensator.

Diesel fuel has a higher energy content per gallon than gasoline, although when measured by the pound it is slightly less. Since the amount of work accomplished by an engine depends on the heat value available in the fuel, more work can be accomplished by burning diesel fuels. The principal characteristics of the common military fuels are shown in Table I-1. Engines operating on the lighter fuels, such as combat gasoline or CITE, experience a significant loss of power output. The use of an automatic fuel-density compensator will increase the flow-rate of the lighter fuels to overcome the loss in power output and will enable the engine to produce its design power output while op-

TABLE I-1

Characteristics of Various Military Fuels

(Approximate values)

Characteristic	Type of fuel							
	Arctic gasoline	Combat gasoline (86/95)	CITE	JP-4	Kerosene	No. 1 Diesel	JP-5	No. 2 Diesel
Specific gravity	0.720	0.730	0.767	0.775	0.805	0.812	0.826	0.849
API gravity	65	62	53	51	44	43	40	35
Heat content, Btu/lb	19,000	19,750	18,400	18,600	18,420	19,330	18,450	19,500
Heat content, Btu/gal	115,000	120,300	117,800	120,300	123,900	131,000	127,200	136,000

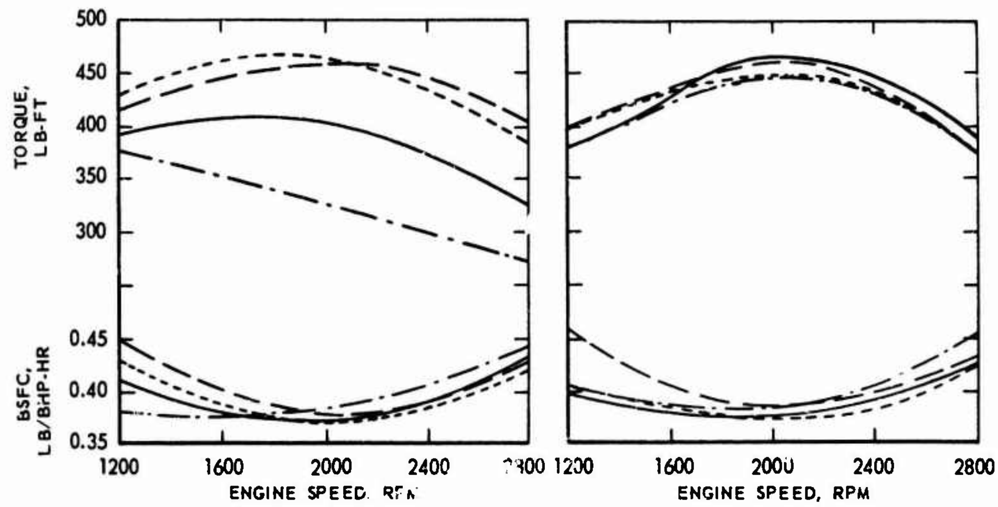


Fig. I-6 —Comparison of Performance Characteristics of a 210-hp Turbosupercharged Compression-Ignition Engine with and without Fuel-Density Compensation

Bsfc, brake specific fuel consumption.

erating on the lighter fuels. Figure I-6 illustrates a comparison of engine performance for an engine with a constant fuel flow-rate and with the fuel flow-rate varied with density. This sensing device can be of either mechanical or electrical design, or a combination of the two. The compression-ignition engine is safer to operate because of the low volatility of diesel fuels and the virtual absence of carbon monoxide in the exhaust.

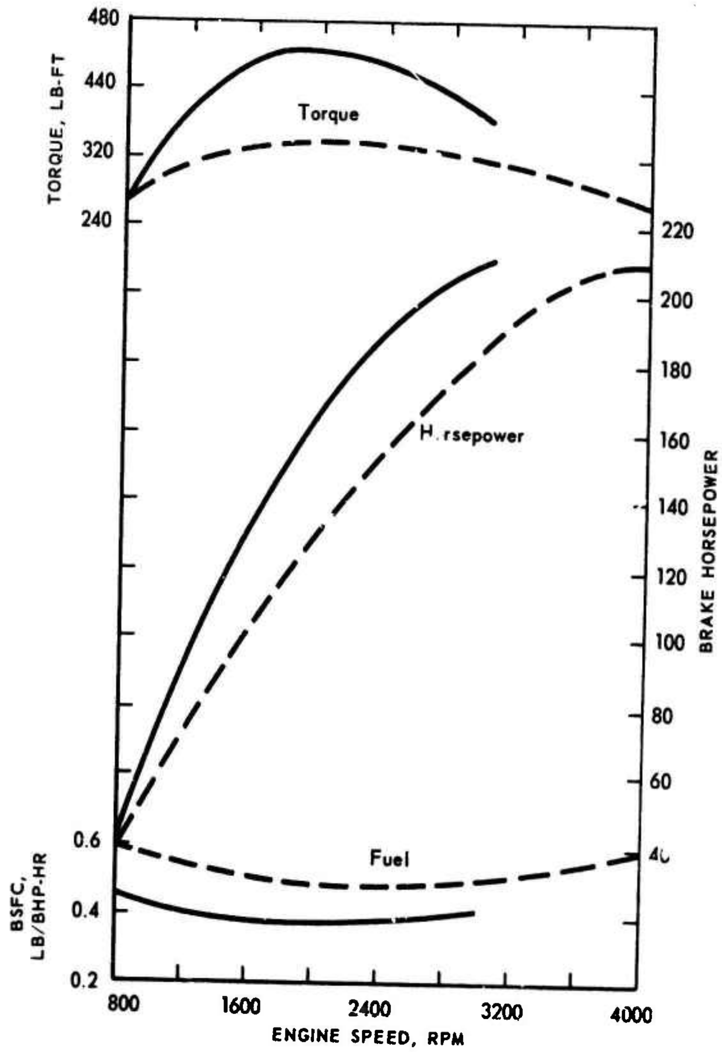


Fig. I-7—Performance and Characteristics Comparison of a Modern Compression-Ignition Engine with a Modern Gasoline Engine

— — — Modern gasoline engine
——— Modern compression-ignition engine

Characteristic	Gasoline engine[a]	Compression-ignition engine[a]
Horsepower	210	210
Weight, lb	630	1150
Volume envelope, ft^3	15.5	18.5
Specific weight, lb/hp	3.0	5.5
Specific output, hp/ft^3	13.6	11.6

[a]Liquid cooled and naturally aspirated.

The performance characteristics of the compression-ignition engine for vehicular application are superior to those of the spark-ignition gasoline engine. The compression-ignition engine develops peak horsepower and torque at lower engine speeds and demonstrates greater fuel economy than the gasoline engine. Figure I-7 illustrates a comparison of the performance characteristics of a compression-ignition engine with those of a spark-ignition gasoline engine of the same power rating. The curves illustrate that the compression-ignition engine has a superiority of approximately 23 percent in torque output, and approximately 26 percent in fuel economy.

Because the compression-ignition engine delivers greater output at lower speed-power ranges, it is more suitably matched for vehicular power and performance than the spark-ignition engine. Because of greater power at lower speed ranges for the compression-ignition engine, less overall gear reduction from engine to wheels or track sprocket is required, and a lower number of intermediate gear ratios is possible.

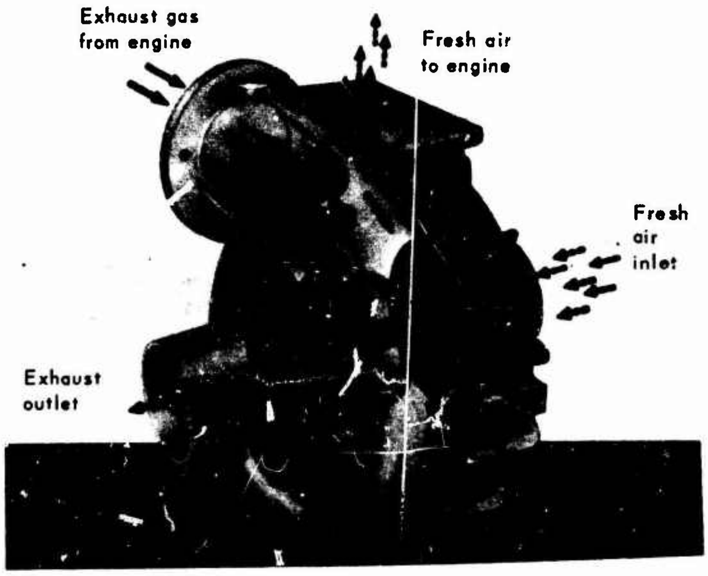


Fig. I-8—Phantom View of a Typical Modern Turbosupercharger

The use of supercharging or turbosupercharging in compression-ignition engines has increased considerably in the past few years. The truck and construction equipment manufacturers have found that their use is an easy and economical method of obtaining higher power outputs. The military has also employed turbosupercharging in both spark-ignition and compression-ignition engines. Supercharging is used to boost the intake pressure of the engine to several times atmospheric pressure, thereby resulting in greater power output. Superchargers used in the past have generally been of the Rootes type or the

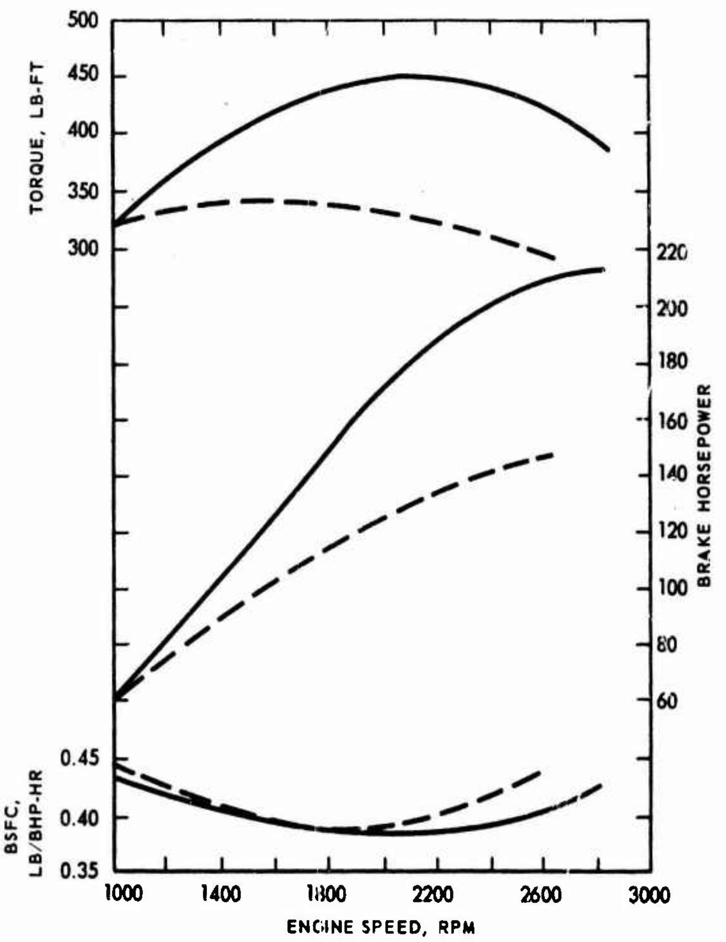


Fig. I-9—Comparison of Compression-Ignition Engine Performance Characteristics Showing Power Increase Due to Turbosupercharging

——— Turbosupercharged — — Naturally aspirated

centrifugal-blower type, both being mechanically driven by the engine. However, the trend has recently been to use turbochargers. A turbocharger differs from a supercharger in that a supercharger is mechanically driven by the engine, whereas a turbocharger is driven by the engine exhaust gases. A turbocharger is in essence a small unfired gas turbine. A typical unit is shown in Fig. I-8. The exhaust gases from the engine drive a radial turbine that is rigidly connected to a shaft and blower. The spent gases, after driving the turbine wheel, are directed out through an exhaust pipe. The blower wheel collects fresh air from the atmosphere and increases its pressure, during the charging cycle, into the engine intake manifold. These units are proved in durability and performance and are readily available commercially. A 25- to 50-percent increase in power output can be realized in some engines, but a 20 to 30 percent increase is considered as a general conservative gain in performance. A comparison of the performance characteristics of a naturally aspirated engine with those of a turbosupercharged engine is shown in Fig. I-9.

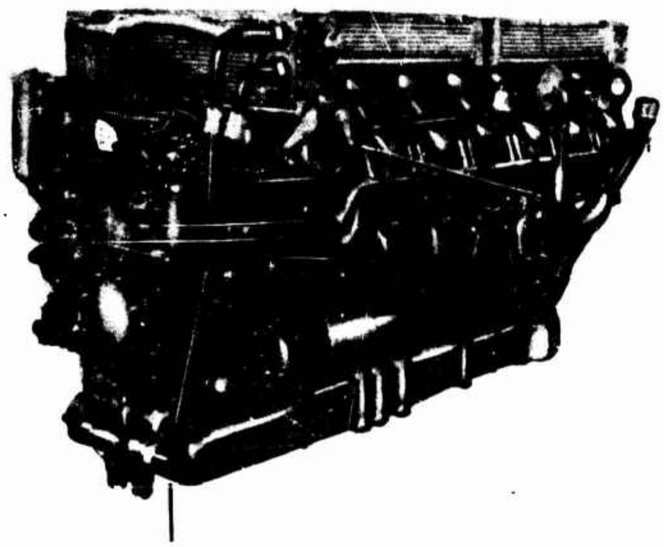
Compression-ignition engines are more reliable and rugged and operate at a higher efficiency than spark-ignition gasoline engines. A modern compression-ignition engine operates at a thermal efficiency approaching 40 percent, whereas the better gasoline engines operate at thermal efficiencies of 28 to 32 percent.

The major disadvantages of the compression-ignition engine are that their weight, volume, and cost are greater than those of a spark-ignition gasoline engine. However, owing to approximately 25 percent better fuel economy the reduced weight and volume of the fuel required in a vehicle with a compression-ignition engine enable it to compete with the spark-ignition engine. With the incorporation of aluminum and the advent of advanced combustion-system technology, the compression-ignition engine will surpass the spark-ignition engine in specific output. Reliability for a combat vehicle is paramount. The compression-ignition engine offers a much wider margin of adaptability than the gasoline engine. And at this point of development the compression-ignition engine is a much more reliable and adaptable power plant than the gas-turbine engine.

SPARK-IGNITION GASOLINE ENGINES

The spark-ignition gasoline engine has been the prime power source for tactical-vehicle propulsion for many years and is still being used today. However, due to the recent military fuel policy, its application has been to small vehicles requiring less than 160 hp. The engines used in tactical vehicles before and during the early part of WWII were modified passenger-car engines that had demonstrated good durability and fuel economy. These engines were used singly in small vehicles and in banks of two or more units for larger tracked vehicles. The liquid-cooled 500-hp V-8 Ford GAA engine was produced in quantity and used as a tank-propulsion unit. This engine was rugged and reliable. The development of a new series of military engines was initiated in 1943. These new engines were based on three cylinder-bore sizes to offer complete family capabilities with a maximum interchangeability of parts and components. These engines were built in 4-cylinder versions of the small-bore size and up to 12-cylinder versions of the large-bore size. Complete power ranges from 127 to 1000 hp were available. The engines were air-cooled and were produced in both opposed-type (AO) series and V-type (AV) series. Some of the engines were fitted with centrifugal-type superchargers for greater power output to fill the power gaps within the series. These engines were lightweight and compact and provided the military with the power-range characteristics they desired. Several of the engines from this family are shown in Figs. I-10 to I-12.

The weight of these engines varied from 4.5 lb/bhp for the smaller engines to 3.0 to 3.5 lb/bhp for the larger tank engines. The specific power output of these engines was 10 to 14 bhp/ft^3 of volume. The fuel consumption of this series was considered good by commercial standards. Figure I-13 compares the performance characteristics of one of the engine series in both naturally aspirated and supercharged versions. The fuel curve shows a minimum, or best-point, specific fuel consumption of approximately 0.48 lb/bhp-hr



AVI-1790-8: 825 hp
AVSI-1790-8: 1000 hp, supercharged

Fig. I-10—Military 12-Cylinder Air-Cooled Spark-Ignition Gasoline Engine

AOI-895-6: 450 hp
AOSI-895-5: 525 hp, supercharged

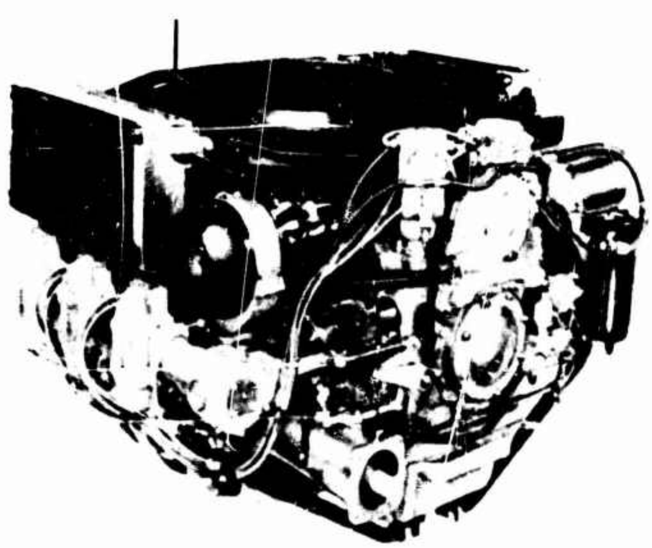


Fig. I-11—Military 6-Cylinder Air-Cooled Spark-Ignition Gasoline Engine

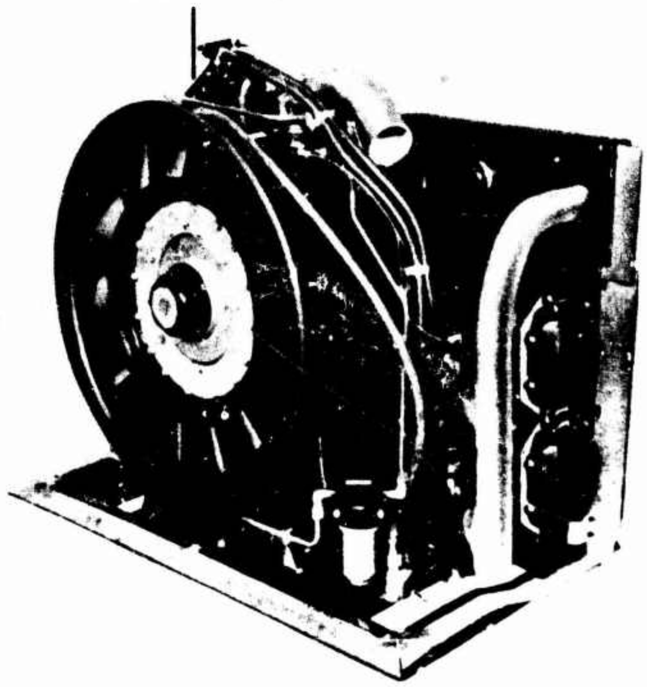


Fig. I-12—Military 4-Cylinder Air-Cooled Spark-Ignition Gasoline Engine

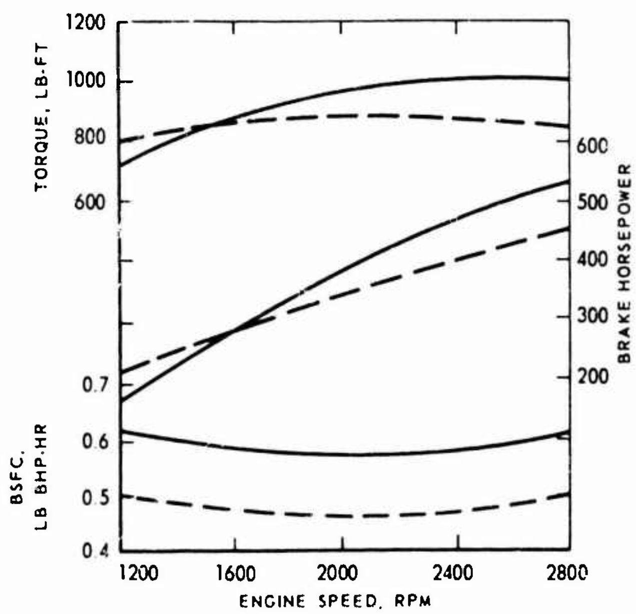


Fig. I-13—Performance Comparison of AOI-895 and AOSI-895 Engines

——— AOSI-895, supercharged

— — AOI-895, unsupercharged

for the naturally aspirated engine and 0.58 lb/bhp-hr for the supercharged version, which is generally representative of all engines in this series.

More recently the Army has developed a small air-cooled gasoline engine to replace the AO-53 engine in the M274-type vehicle. This engine, the AO-42 shown in Fig. I-14, is based (as was the AO-53 engine) on one of a series of small industrial engines developed for the Army. The AO-42 is a 14-hp 4-stroke-cycle 2-cylinder opposed-piston air-cooled spark-ignition gasoline engine. This unit weighs 152 lb dry and has a volume of 6.6 ft^3. The specific weight is 10.7 lb/bhp and the specific power output is 2.22 bhp/ft^3, which is considered poor. The fuel consumption of the AO-42 is approximately 0.65 lb/bhp-hr, which is considered very good for an engine in this power class.

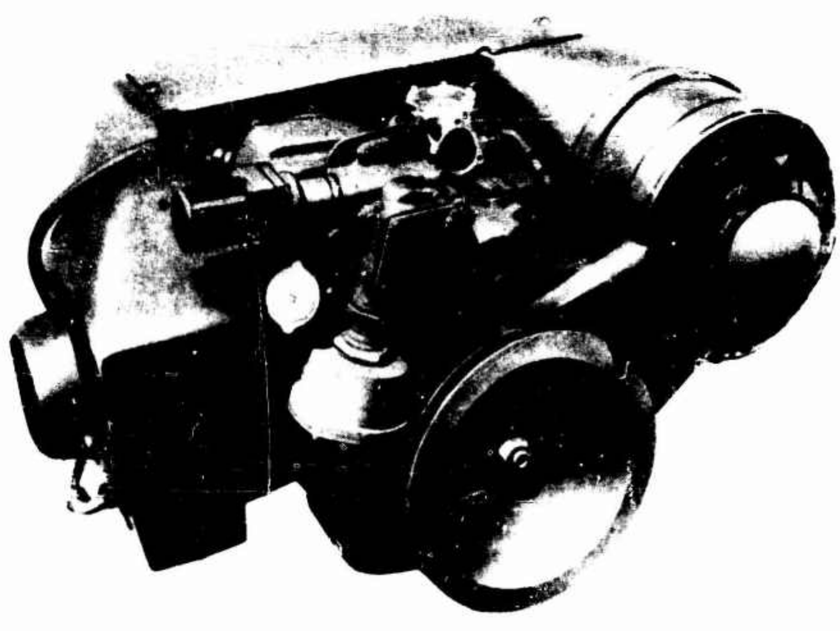


Fig. I-14—AO-42 Engine

Some recent work on improving small engines of this type has been conducted by the Southwest Research Institute. They investigated the possibility of achieving higher power outputs from the 4AO32 engine and reducing its fuel consumption. This is a small air-cooled spark-ignition gasoline engine that produces a maximum of 9.75 bhp. The minimum fuel consumption of the standard engine was 0.82 lb/bhp-hr. Their investigation[1] showed that by modifying the breathing and combustion characteristics an increase in power output of 77 percent and a decrease in fuel consumption of 19 percent was achieved. The corresponding volumetric efficiency increase was 41 percent. The higher power output was partly accomplished by accepting a reduction of engine life from 1500 to 1000 hr.

The L-141 engine was designed specifically for and used in the M151 ¼-ton utility truck. This unit is a 4-cycle 4-cylinder in-line liquid-cooled spark-ignition gasoline engine that delivers 71 hp. It is a rugged and well-constructed unit. The performance characteristics of this engine are shown in Fig. I-15. The fuel curve shows a gross specific fuel consumption of 0.46 lb/bhp-hr, which is excellent for an engine of this power size. It is considered representative of the best technology relating to fuel economy for spark-ignition gasoline engines.

To provide power for the smaller tracked and special-purpose vehicles the Army has utilized several well-known rugged and reliable commercial spark-ignition passenger-car and truck engines. The major advantage of adapting proved commercial engines is that of economy and availability. The military does not have the burden of developing, perfecting, or tooling the engine for mass production. Rather elaborate commercial facilities exist, which are necessary to manufacture large quantities of engines. Because these facilities exist, the delay usually associated with the mass production of certain high-density engines during full mobilization is minimized.

Commercial engines must be modified to adapt them to military use. The engines must be derated from the commercial rating practice to provide the durability and life required for military engines. Military engines operate in poor environments at high power loadings, as compared to the family passenger car or light truck. As an example, a typical modern passenger-car gasoline engine rated at 220 to 240 hp must be derated to between 160 and 180 hp for military-vehicle application. In addition, it is necessary to:

(a) Incorporate an electrically shielded and waterproof ignition system.

(b) Incorporate a 24-volt high-output generating and starting system.

(c) Use a deep oil-sump pan for slope operation.

(d) Waterproof the engine.

(e) Incorporate necessary accessory power takeoffs.

(f) Incorporate a carburetor to meet military operating characteristics, which include slope operations.

A typical commercial engine adapted for use in military vehicles is the Chevrolet 283 engine. The 283 is a 4-stroke-cycle V-8-cylinder spark-ignition liquid-cooled naturally aspirated gasoline engine. It has a bore of 3.875 in. and a stroke of 3.000 in., with a total piston displacement of 283 $in.^3$ It is militarily rated from 160 to 175 gross bhp at 4400 rpm. The engine has a dry weight of 531 lb and is approximately 28 in. long, 25 in. wide, and 32 in. high, for a volume envelope of approximately 13.5 ft^3. The specific weight of the 283 engine (at 160 bhp) is 3.3 lb/bhp and the specific power output is 12 bhp/ft^3 of volume. The performance characteristics of the 283 engine are shown in Fig. I-16. The minimum, or "best-point," fuel rate is 0.52 to 0.53 lb/bhp-hr.

Recently the United States Army Tank-Automotive Center (USATAC) has proposed to build a 100-hp gasoline engine based on the cylinder assemblies from the AO-53 engine. This engine is intended to fill what is considered a gap in this power range, for application to vehicles such as the XM561 and XM571. The proposed AO-198 unit is a 4-cycle naturally aspirated 8-cylinder opposed-piston spark-ignition gasoline engine. It would develop 100 bhp at 4600 rpm. The total weight is estimated at approximately 350 lb and the envelope volume is approximately 15.5 $ft.^3$ The specific weight of this unit is approximately 3.5 lb/bhp and the specific power output is approximately 6.5 bhp/ft^3 of volume. The specific fuel consumption of the AO-198 engine will be in the region of 0.6 lb/

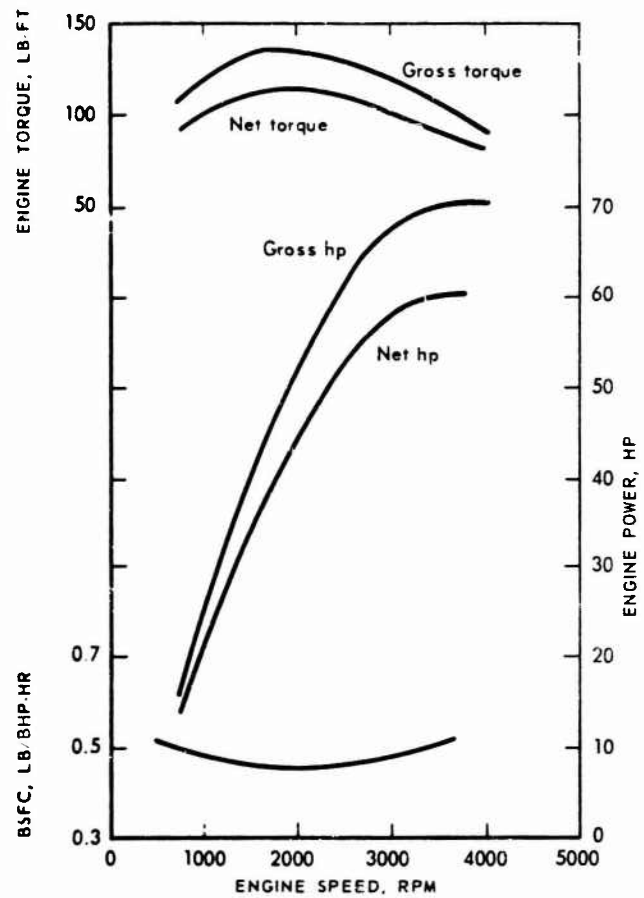


Fig. I-15—Performance Characteristics of L-141 Spark-Ignition Gasoline Engine

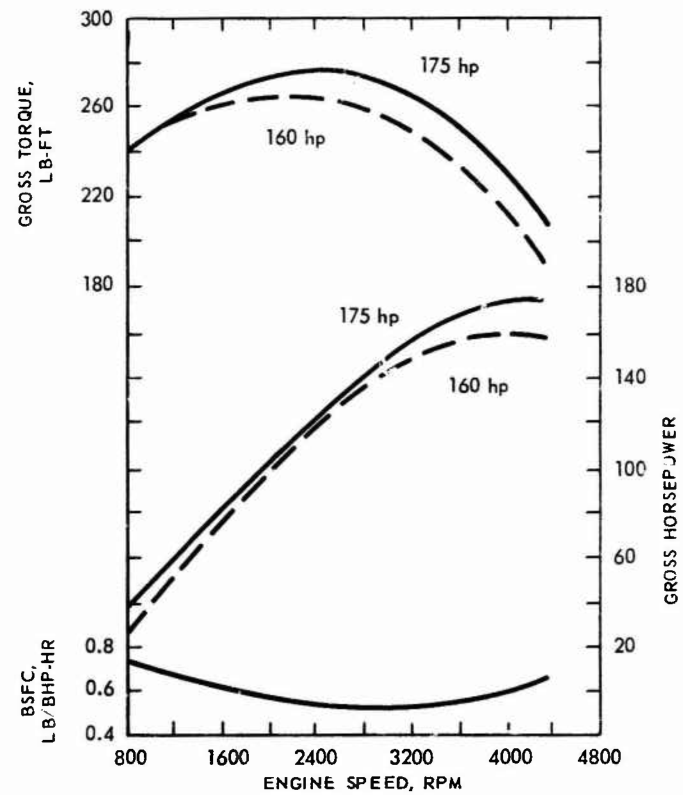


Fig. I-16—Performance Characteristics of 283 Engine at 160- and 175-hp Rating

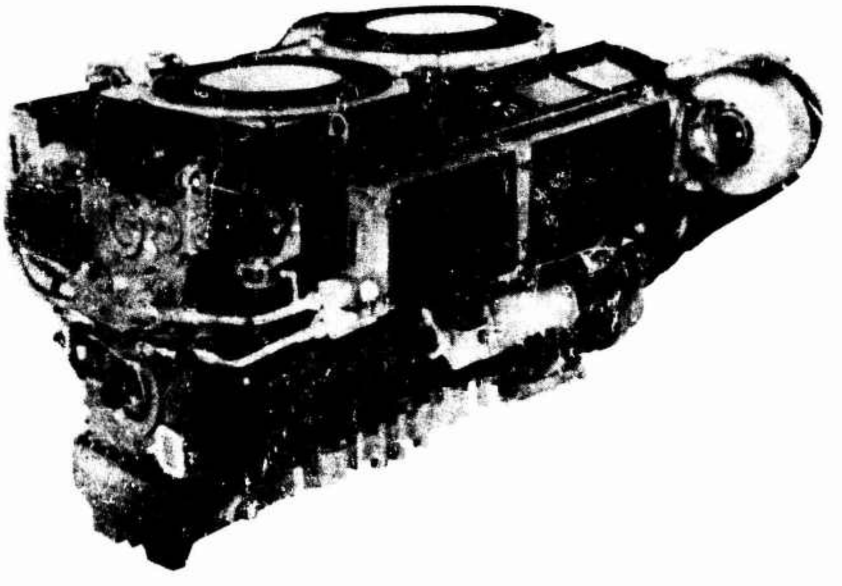


Fig. I-17—AVDS-1790-2 Compression-Ignition Tank Engine

SPECIFICATIONS

Displacement, in.^{3}	1790
Number of cylinders	12
Cylinder arrangement	90-deg upright V
Bore, in.	5.750
Stroke, in.	5.750
Rated speed, rpm	2400
Idle speed	750
Rated bhp	750
Rated BMEP, psi	139
Compression ratio	16:1
Supercharging	Turbo
Cycle	4
Cooling	Air
Length, in.	70.6
Width, in.	75.2
Height, in.	44.3
Volume, ft^3	136
Weight (lb) dry (with accessories)	4527
Volume, hp/ft^3	5.5
Specific weight, lb/hp	6.0

bhp-hr, which is considered poor. A turbosupercharged version of the AC-198 engine would have an output of 125 to 130 hp. The specific weight for the turbosupercharged version would be approximately 2.8 to 3.0 lb/bhp, and the specific power output would be approximately 12.5 bhp/ft^3 of volume.

COMPRESSION-IGNITION ENGINES

Conventional Compression-Ignition Engines

Conventional compression-ignition engines are those that do not incorporate any radical design departure from that commonly known as the "diesel engine." The incorporation of supercharging or turbosupercharging, precombustion chambers, recessed-cup piston crowns, high compression ratios, 2-cycle or 4-cycle concepts, air or liquid cooling, or the ability to operate on more than one type of fuel does not classify the power source as other than conventional. This applies to both military developed engines and commercially available engines.

Military Compression-Ignition Engines

The AVDS-1790 engine was the first fully successful military-sponsored compression-ignition engine. This engine resulted from an attempt to convert an Ordnance Corps AV-1790 series V-12-cylinder spark-ignition gasoline engine to the diesel cycle. This program, initiated in 1954, resulted in the successful development and production of the AVDS-1790-2 compression-ignition engine, shown in Fig. I-17. It was incorporated into the M60 105-mm-gun main battle tank. The performance characteristics of the AVDS-1790-2 engine are shown in Fig. I-18. The AVDS-1790-2 engine can operate on diesel fuel or,

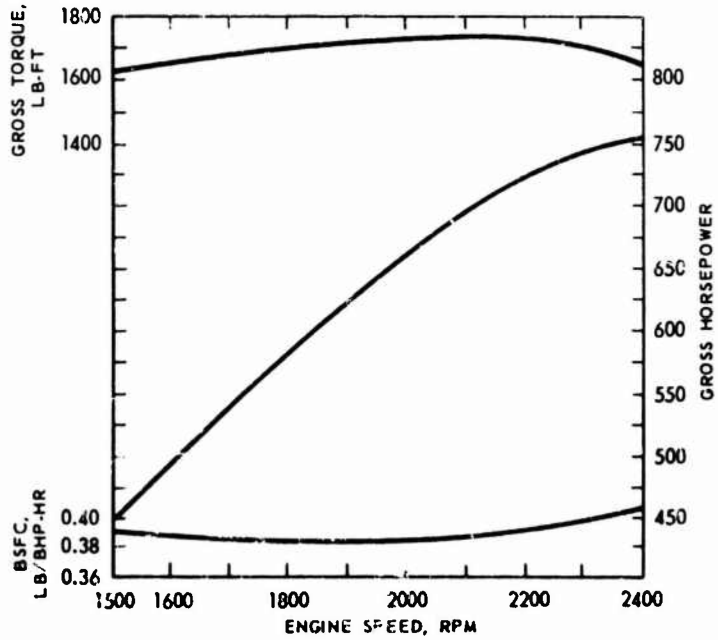


Fig. I-18—Performance Characteristics of a 750-hp AVDS-1790-2 Compression-Ignition Tank Engine with limitations, on CITE fuel. The engine has a specific weight of 6.0 lb/hp and a specific output of 5.5 hp/ft^3 of volume. These specifications could be improved by today's technology. The minimum fuel consumption of the AVDS-1790-2 engine is 0.385 to 0.390 lb/bhp-hr, which is considered good.

A program was initiated with the Continental Motors Corporation in 1955 for the development of a multifuel engine for use in cargo trucks and small tracked and special-purpose vehicles. The design of this engine was based on the patented MAN or hypercycle combustion principle. A cutaway view of the hypercycle system design is shown in Fig. I-19. This system differs from the

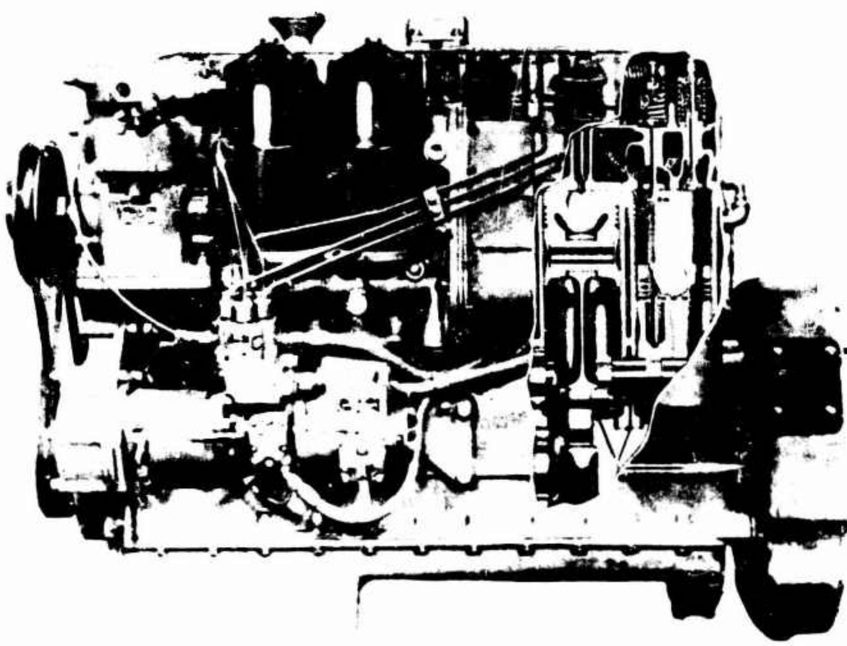


Fig. I-19—Hypercycle Combustion System Design

more common direct-injection compression-ignition engine combustion system in that each cylinder incorporates a hemispherical combustion chamber in the crown of its piston. A fuel injector protrudes from the cylinder head into the recessed combustion chamber in the piston. A spiral intake port imparts a vigorous swirl to the intake air. The principle of operation of the hypercycle multifuel combustion system is illustrated in Fig. I-20. The ignition of gasoline by the heat of compression in a conventional diesel-cycle engine causes excessively rapid increases in pressure. This is avoided in the hypercycle system by depositing all but a small portion of the injected fuel as a thin film on the walls of the combustion chamber. The fuel burns slowly as the rapidly

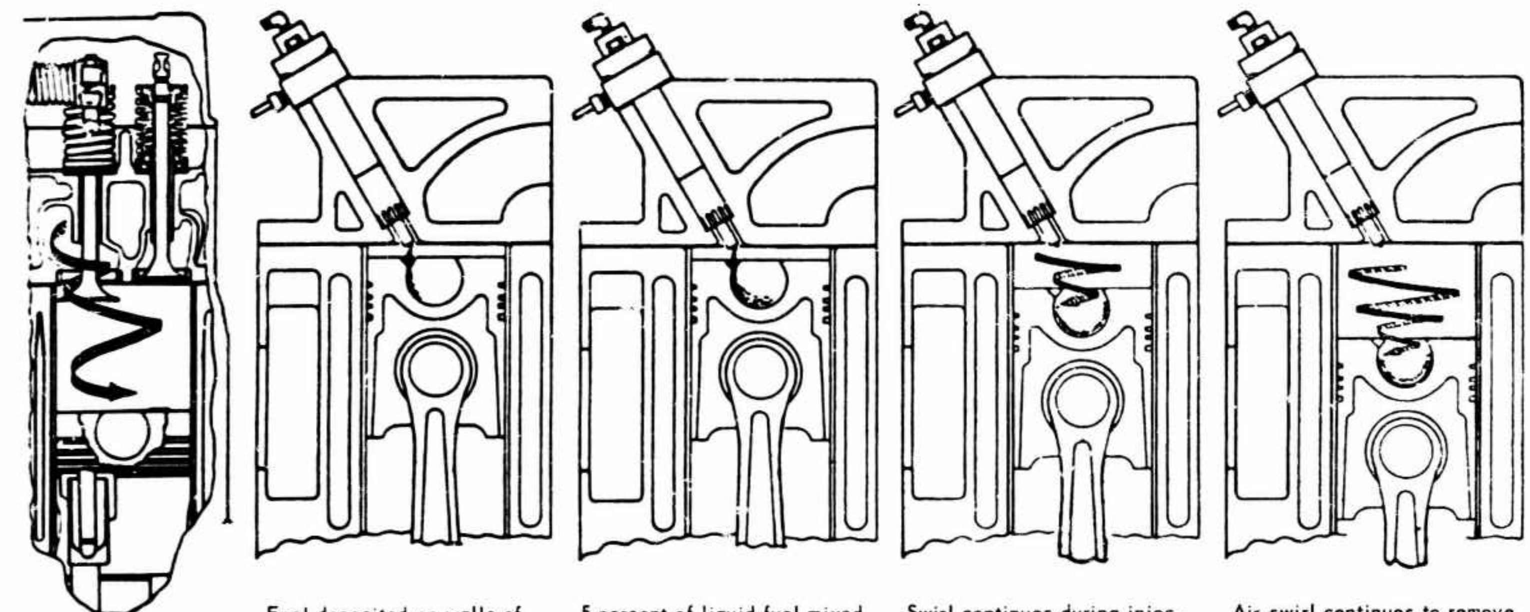
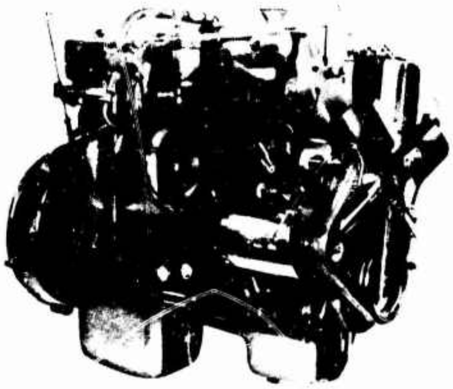
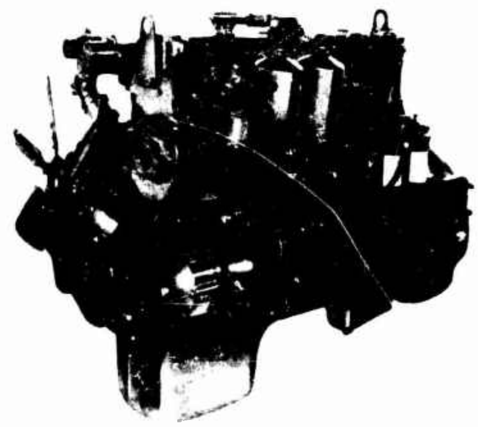


Fig. I-20—Principle of Operation of the Hypercycle Multifuel Combustion System



LDS-465-1:
210 hp



LD-465-1:
140 hp

Fig. I-21—Hypercycle Multifuel Engine

SPECIFICATIONS

Model	LD-465-1	LDS-465-1
Type	Multifuel	
Cycle	4-stroke	
Coolant	Liquid	
Fuel	Diesel, cite, gasoline	
Cylinders	6-in. line	
Bore and stroke	4.56 in. × 4.87 in.	
Displacement	478 cu. in.	
Compression ratio	22:1	
Rated power (all fuels)	140 bhp 2600 rpm	210 bhp 2800 rpm
Rated torque	330 lb-ft 1800 rpm	450 lb-ft 2000 rpm
BMEP	90 psi 2600 rpm	125 psi 2800 rpm
Aspiration	Natural	Turbosupercharger
Dimensions L · W · H	48 in. · 29 in. · 40 in.	
Volume	32 cu. ft	
Weight, dry (w all accessories)	1500 lb	1561 lb
Specific output	4.4 bhp ft^3	6.5 bhp ft^3
Specific weight	10.7 lbs bhp	7.4 lbs bhp

swirling air removes it from the hemispherical combustion chamber and thoroughly mixes with it. This combustion system, by controlling the rapid pressure rise, enables these engines to operate on many grades of fuel.

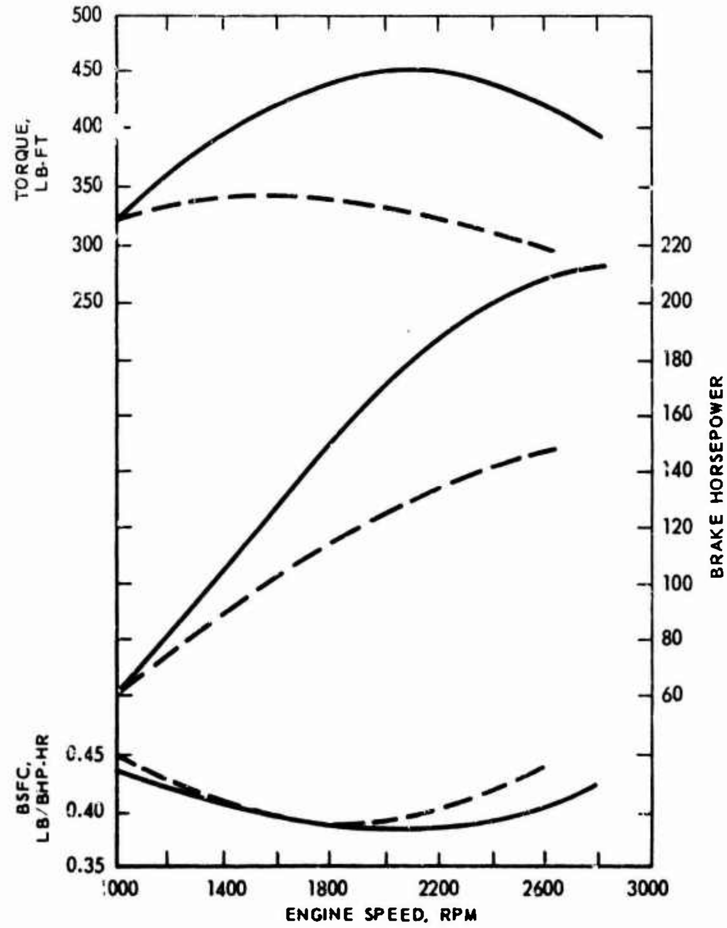


Fig. I-22—Performance Characteristics of Hypercycle Multifuel Engines

LDS-465-1, turbosupercharged

LD-465-1, naturally aspirated

Successful development by the US Army resulted in the production of the hypercycle engine in two sizes for the $2\frac{1}{2}$- and 5-ton cargo truck. The basic engine is the 140 hp naturally aspirated LD-465-1, for use in the $2\frac{1}{2}$-ton truck. A turbosupercharged version of this engine is the 210 hp LDS-465-1, for use in the 5-ton truck. These engines are illustrated in Fig. I-21, and their performance characteristics are shown in Fig. I-22. The LD-465-1 engine has a specific weight of 10.7 lb/hp and a specific output of 4.4 hp/ft^3 of volume. The turbocharged LDS-465-1 has a specific weight of 7.4 lb/hp and a specific output of 6.5 hp/ft^3 of volume. These specifications are poor by today's technology. However, the use of aluminum construction in place of cast iron would reduce the weight by approximately 20 percent. The minimum fuel consumption of both engines is 0.38 lb/bhp-hr, which is considered good. The hypercycle engines have a potential of increasing power by 20 to 25 percent through further development.

Commercial Compression-Ignition Engines

The military has in the past utilized commercial compression-ignition engines where military-designed engines in required power ranges were not available. These engines have been supplied by several prominent engine manufacturers and have demonstrated good durability. They have seen world-wide dependable service in industrial, automotive, and marine applications. One such power plant is the V8-300, manufactured by the Cummins Engine Company, which is used in the LARC-V and LARC-XV amphibious vehicles. The V8-300 is a 4-cycle V-8-cylinder liquid-cooled compression-ignition engine.

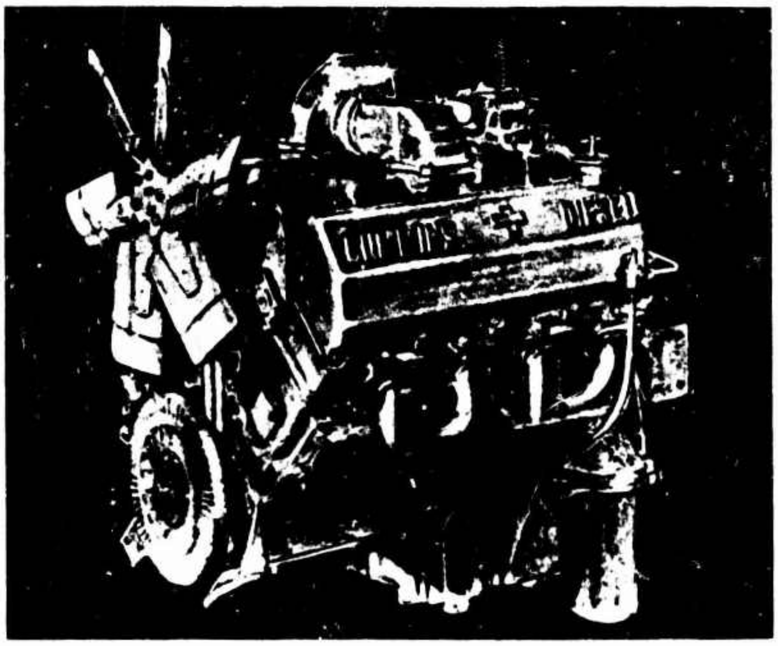


Fig. I-23—V8-300, 8-Cylinder, 4-Cycle, 300-hp Compression-Ignition Engine Used in Military Vehicles

The V8-300, shown in Fig. I-23, develops 300 hp at 3000 rpm with natural aspiration and 380 to 400 hp if turbosupercharged. The complete unit weighs approximately 1800 lb complete. This weight could be reduced to 1450 lb by the use of aluminum instead of cast iron. The volume of this engine is 31 ft^3. The specific weights and power outputs of the V8-300 are shown in Table I-2. The minimum fuel consumption of the V8-300 engine is 0.380 lb/bhp-hr, which is comparable to other modern engines of its power rating. Other engines manufactured by this company also possess similar size, weight, and operating characteristics.

The most prominent commercial compression-ignition engines used in military vehicles are those manufactured by the Detroit Diesel Division of

GMC. These engines are representative of the best commercially developed units offering light weight, high performance, rugged dependability, and reasonable cost. The basic models 53, 71, and 149 series engines (the figures denote piston displacement in cubic inches per cylinder) are available in various power ratings and in 2-, 3-, 4-, and 6-cylinder in-line versions and 6-, 8-, 12-, and 16-cylinder Vee-type versions.

TABLE I-2

Primary Specifications of Commercial V8-300 Compression-Ignition Engine

Item	Naturally aspirated		Turbosupercharged	
Brake horsepower	300	300	400	400
Construction	Cast iron	Aluminum	Cast iron	Aluminum
Weight, lb	1800	1450	1850	1500
Volume, ft^3	31	31	32	32
Specific weight, lb/bhp	5.0	4.8	4.6	3.8
Specific weight, bhp/ft^3	9.7	9.7	12.5	12.5

Within each series, all engines, regardless of horsepower or number of cylinders, contain the same basic design and construction, the same internal working and moving parts (with the exception of crankshaft and camshaft), and the same external components. The interchangeability of parts considerably reduces logistic support requirements. These engines are liquid-cooled, operate on the 2-stroke cycle, and incorporate an open-type combustion chamber with uniflow scavenging supplied by a mechanically driven Rootes-type blower. The engines use air-inlet porting and conventional poppet-type exhaust valves. Figure I-24 illustrates a cutaway view of the GMC engine-cylinder design. These engines have a compression ratio of 17 to 1.

In 1961 new developments were introduced that could adapt these engines for multifuel operation with only minor modifications to the standard production models. The modifications include a new piston design with a modified bowl in its crown to increase the compression ratio from 17:1 to 23:1 in order to ignite fuels with a lower specific gravity, such as combat gasoline. A new needle-valve fuel injector and a higher-capacity fuel-transfer pump are also used. Existing standard engines can be converted in the field to multifuel engines by incorporation of these parts. Since a higher compression ratio is required for multifuel capabilities the life of some standard components is reduced. However, several test engines incorporating the above modifications were tested and found to have excellent durability.

The GMC series compression-ignition engines are available for military use from approximately 60 to 1200 hp in either cast-iron or aluminum construction. All these engines are adaptable to turbosupercharging. A typical GMC engine, the 6V-53 rated at 210 hp, is shown in Fig. I-25. This unit is used in the M113A1 armored personnel carrier.

The primary specifications of the GMC series engines are shown in Table I-3. The specific weight of the 6V-53 turbocharged engine in aluminum construction is 3.6 lb/bhp, and the specific output is 10.7 bhp/ft^3 of volume.

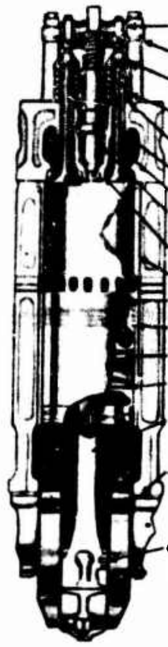


Fig. 1-24—Cutaway View of 53, 71, and 149 Series Engine Cylinder

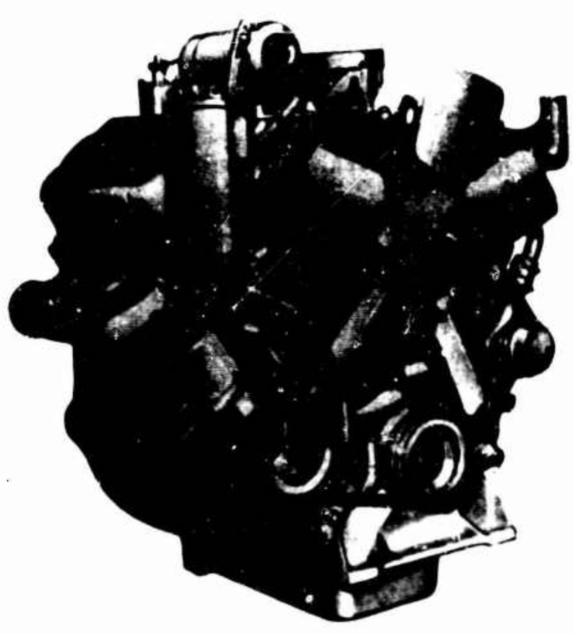


Fig. 1-25—6V-53 6-Cylinder 2-Cycle 210-hp Compression-Ignition Engine Used in Military Vehicles

The specific weights of these engines are the highest available commercially and are equal to or better in some power classes than current military-developed engines. The volume of these engines compares with the best commercially available engines, but when compared to current military-developed engines the specific power output per cubic feet of volume is low. The minimum fuel consumption of the GMC series engines varies from 0.37 to 0.42 lb/bhp-hr, depending on engine horsepower size, which is comparable to other modern compression-ignition engines. These engines are representative of the best commercial-engine technology available today.

TABLE I-3

Primary Specifications of Several GMC Series Engines[a]

Item	Engine model				
	3-53	6V-53	8V-71	12V-71	8V-149
Brake horsepower					
Naturally aspirated	103	220	350	525	—
Turbosupercharged	—	300	to 530	to 800	to 960
Construction	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum
Weight (dry), lb					
Naturally aspirated	750	1050	1540	2100	—
Turbosupercharged	—	1090	1710	2300	3300
Volume, ft^3 (approx.)	18	28	37	52	58
Specific weight, lb/bhp					
Naturally aspirated	7.3	5.0	4.4	4.0	—
Turbosupercharged	—	3.6	—	2.9	3.45
Specific output, bhp/ft^3					
Naturally aspirated	5.7	7.8	9.5	10.1	—
Turbosupercharged	—	10.7	14.4	15.4	16.5

[a]Weight, volume, and power values may vary, depending on application and duty cycle. Values shown are maximum.

Perkins Engine Inc. is currently developing a differentially supercharged compression-ignition engine-transmission in an effort to provide desirable low-speed performance characteristics with a minimum of complexity. The differentially supercharged diesel engine is a complete engine-transmission unit. This system consists of a compression-ignition engine, a mechanically driven supercharger, and a torque converter with either a manual or automatic shift transmission. A schematic diagram of this concept is illustrated in Figs. I-26 and I-27. The engine is supercharged by a positive-displacement blower that is mechanically (and differentially) driven in such a way that the charge density (and therefore the output torque) increases with decreasing engine speed. The engine-transmission system must be properly matched to obtain optimum performance characteristics. Figure I-28 illustrates a truck-performance comparison[2] between a differentially supercharged diesel engine with a 2-speed transmission and a conventional supercharged diesel engine with a 5-speed transmission in a 16-ton gross-vehicle-weight (GVW) vehicle. The tractive-effort curves illustrate the superior performance of the differentially supercharged

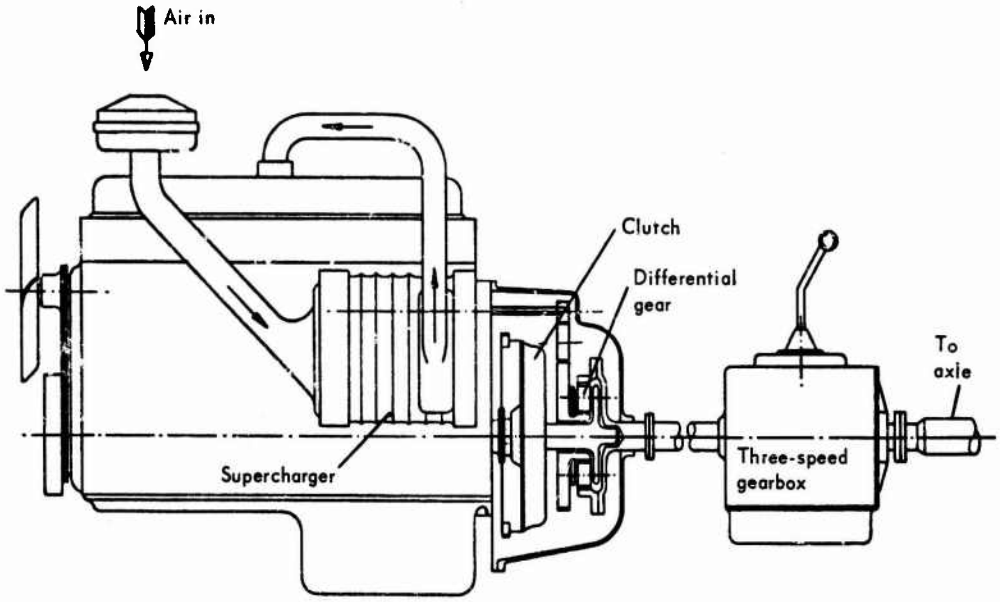


Fig. 1-26—Simple Differential-Supercharging Arrangement

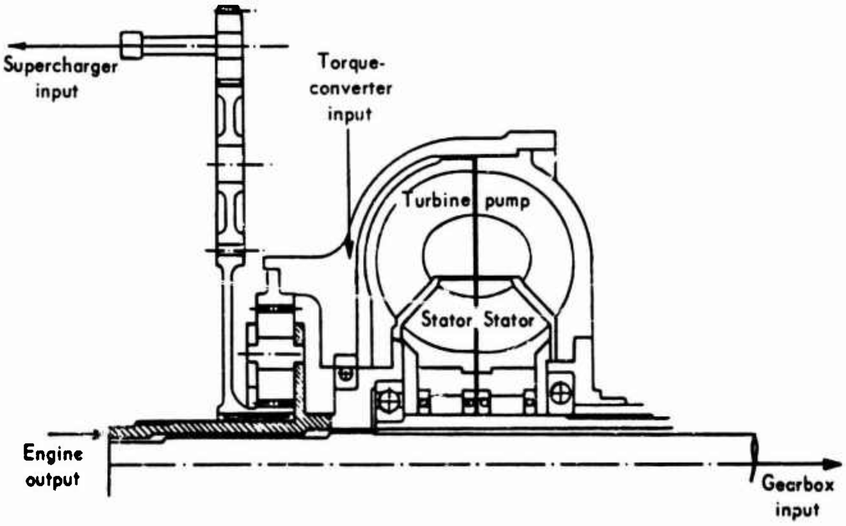


Fig. 1-27—Combined Differential and Torque-Converter Arrangement

engine in all speed ranges and particularly at lower speeds. This system has better low-speed performance characteristics and a much simpler transmission and requires less effort and skill by the operator. The differential-supercharged concept can be applied to any reciprocating engine and to most transmissions for both commercial and military use.

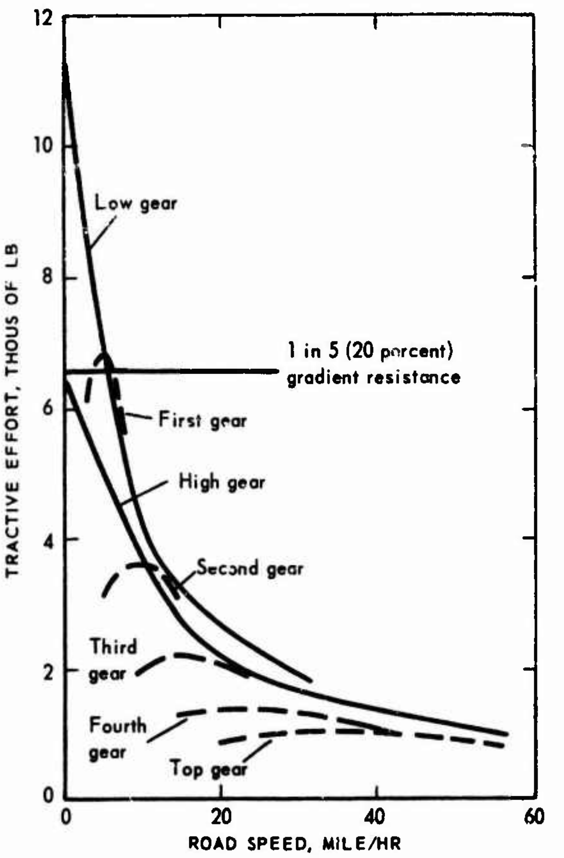


Fig. I-28—Performance-Characteristic Comparison between Differentially Supercharged Diesel Engine with 2-Speed Transmission and Conventionally Supercharged Diesel Engine with 5-Speed Transmission, in a 16-ton GVW Truck[2]

——— Differential engine
— — Conventional engine and transmission

AVM-Series Engines

In 1952 the Ordnance Tank-Automotive Command (OTAC; now ATAC), investigated the possibilities of developing a lightweight and compact air-cooled compression-ignition engine based on the 2-stroke-cycle loop-scavenged principle. After analyzing concept studies submitted by industry, OTAC initiated a research and development program in 1954, which was contracted to the Lycoming Division of AVCO Corporation. Initial development and testing were on

1- and 2-cylinder test engines. Development and test work were conducted on a limited scale until 1961, when work was started on the 4-cylinder full-sized engine. In 1963 a requirement for multifuel capabilities was incorporated into the development of the AVM engine.

The AVM series air-cooled Vee-type multifuel engines are designed around the basic cylinder assembly shown in Fig. I-29. These engines operate on the 2-stroke-cycle principle and employ loop scavenging. The AVM engine operates at an effective compression ratio of 18.7 to 1. It incorporates an open-type combustion chamber, direct fuel injection, an omega-cup configuration in the top of the piston, and automatic throttling of intake air at part load, as shown in Fig. I-30.

The multifuel capability of the engine is good owing to the compact heat-conserving combustion chamber, the lack of chilling valves in the head, throttling of the air intake at part loads, and an effective compression ratio. The original cylinder assembly was made of aluminum alloy and constructed with an integral head that eliminates the need for gaskets and bolts. This cylinder was hard chrome plated and eliminated the need for a separate liner. However, the later engines utilize a nitride steel liner that can be pressed into the cylinder or inserted during the cylinder-casting process to form an integral unit. The pistons rotate to carry heat from the hot exhaust-port region to the cool intake-port region. In addition, the pistons are oil-cooled by stationary oil jets that direct cooling oil to the piston crown. Rotation of the piston is accomplished by a unique design—upper connecting rod, barrel-type wrist pin, and two-piece piston assembly. The engines are valveless and have a total of 24 moving parts per cylinder. Ease of accessibility of all parts facilitates their servicing or replacement. A cylinder assembly complete with piston and connecting rod can be removed and replaced in approximately 45 min without removal of engine from the vehicle. Scavenging of the cylinders is accomplished by a gear-driven centrifugal blower located at the front of the engine.

The AVM engine has good cold-starting characteristics. Starting is aided by a glow plug so located in the cylinder head that the fuel spray impinges on its tip.

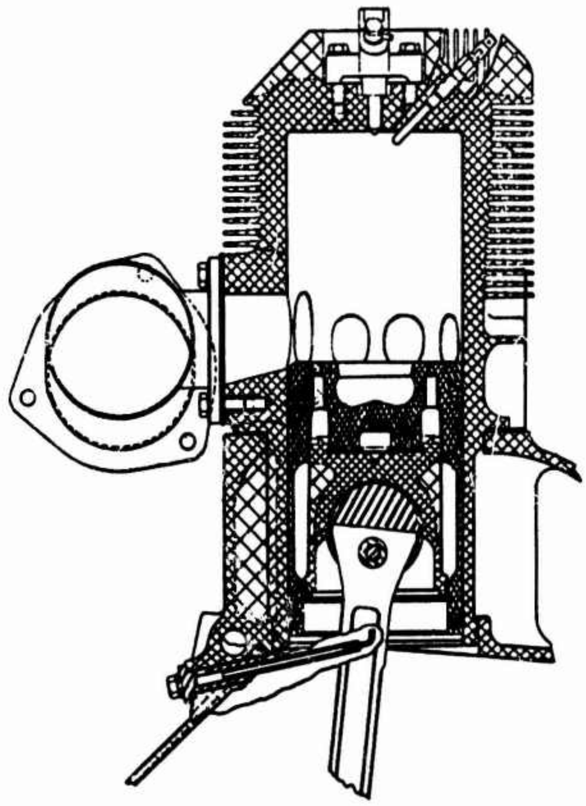
The major components, such as the crankcase, cylinder, and piston assemblies, are constructed of aluminum to achieve low weight.

Approximately 35 AVM-310 prototype engines have been built to date. These engines have completed several thousand hours of dynamometer testing, and several engines have been installed in small developmental wheeled vehicles for operational and durability testing. Three prototype AVM-625 engines have been constructed and are currently undergoing dynamometer testing. The first prototype AVM-470 engine is being constructed and dynamometer testing was anticipated to begin in the late fall of 1966.

The AVM series family of engines is intended for use in light tracked and wheeled vehicles. These engines could have application in the newly developing 2½- and 5-ton cargo trucks.

Discussion. The AVM series engines are of Vee-type configuration. Front and rear views of the AVM-310 engine are shown in Fig. I-31. The AVM-310 engine is one member of a possible family of engines built around the same bore size by varying the number of cylinders. The family capabilities and characteristics of AVM series engines are shown in Table I-4.

Fig. I-29—Cross Section of AVM Cylinder and Piston Assemblies



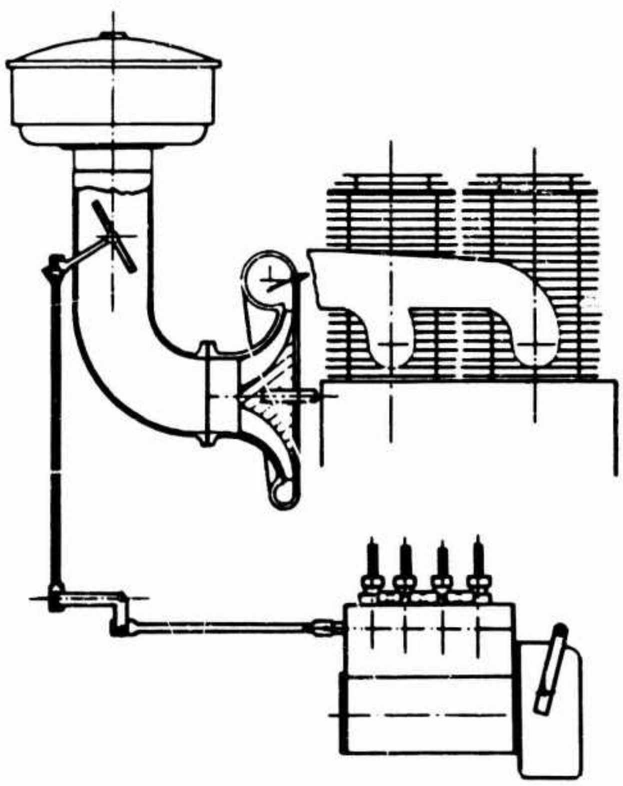
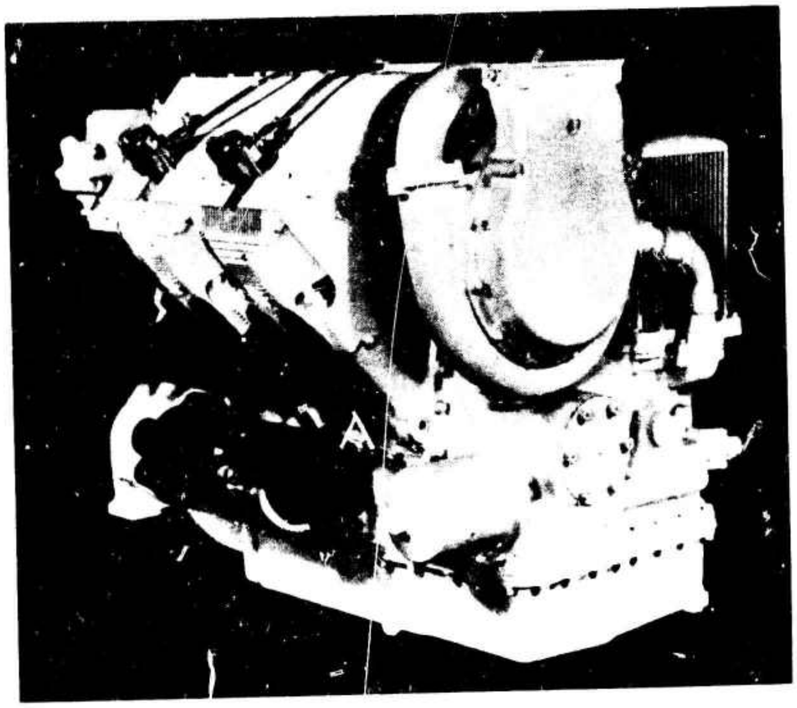
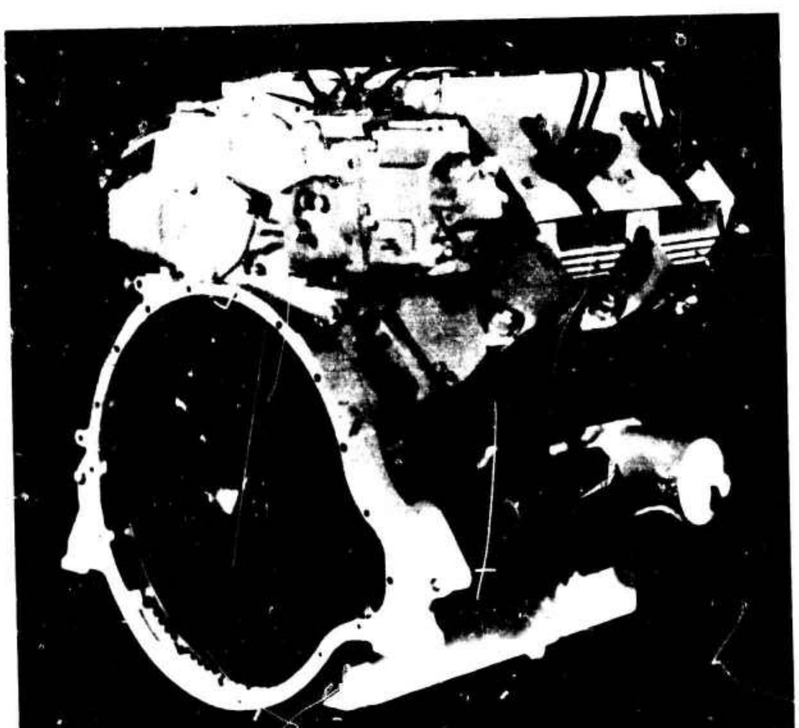


Fig. I-30—Automatic Throttling of Intake Air by Linkage to Fuel Rack



a Front View



b. Flywheel View

Fig. I-31—AVM-310 Engine Assembly

The AVM engines are air-cooled. A suction-type cooling blower located on top of the engine draws air past the cylinders and oil cooler. The cylinders are finned and are effectively shrouded to direct the air flow with minimum pressure drop and leakage loss. A maximum use of common parts and components is made for all engines of the family. The AVM engines can be manufactured by use of present automotive engine tooling since the power-producing components are similar to the reciprocating components of more conventional compression-ignition engines.

TABLE I-4

Characteristics of AVM Series Engines

Item	Model		
	AVM-310	AVM-470	AVM-625
Type and no. of cylinders	V-4	V-6	V-8
Bore and stroke, in.	4.25 × 5.5	4.25 × 5.5	4.25 × 5.5
Displacement, in.^{3}	312	468	624
Compression ratio	18.7:1	18.7:1	18.7:1
Brake horsepower, gross, at 260 rpm	135	215	280
Brake horsepower, net, at 2600 rpm	120	180	250
Minimum fuel consumption, gross, lb/bhp-hr	0.425	0.425	0.425
Fuel consumption at rated power, gross, lb/bhp-hr	0.440	0.440	0.440
Dimensions, in. L × W × H	29 × 30 × 29	37 × 30 × 29	45 × 30 × 29
Volume, ft^3	14.4	18.6	22.6
Weight, lb (dry complete unit-cooled engine)	690	890	1120
Specific output, gross, bhp/ft^3	9.4	11.5	12.5
Specific weight, gross, lb/bhp	5.10	4.15	4.00

The specific weight of the AVM series varies from 5.1 (AVM-310) to 4.0 lb/bhp (AVM-625), with specific power outputs of 9.4 to 12.5 bhp/ft^3 of volume. The weight and volume of the AVM series engines are high when compared to other high output compression-ignition engines, but they are considerably

Fig. I-32—Physical Comparison of AVM-310 Engine with Modern Truck Diesel Engine

AVM-310, 120 net bhp, 670 lb, 14.4 ft^3

Modern truck diesel, 112 net bhp, 1100 lb, 20 ft^3

lighter and more compact than conventional truck diesels, as shown by the comparison in Fig. I-32. It must be realized that when a comparison is made for specific outputs of a liquid-cooled engine with those of an air-cooled engine, the weight and volume allowance of the cooling system must be considered. Air-cooled engines with integral cooling systems are lighter and less bulky than

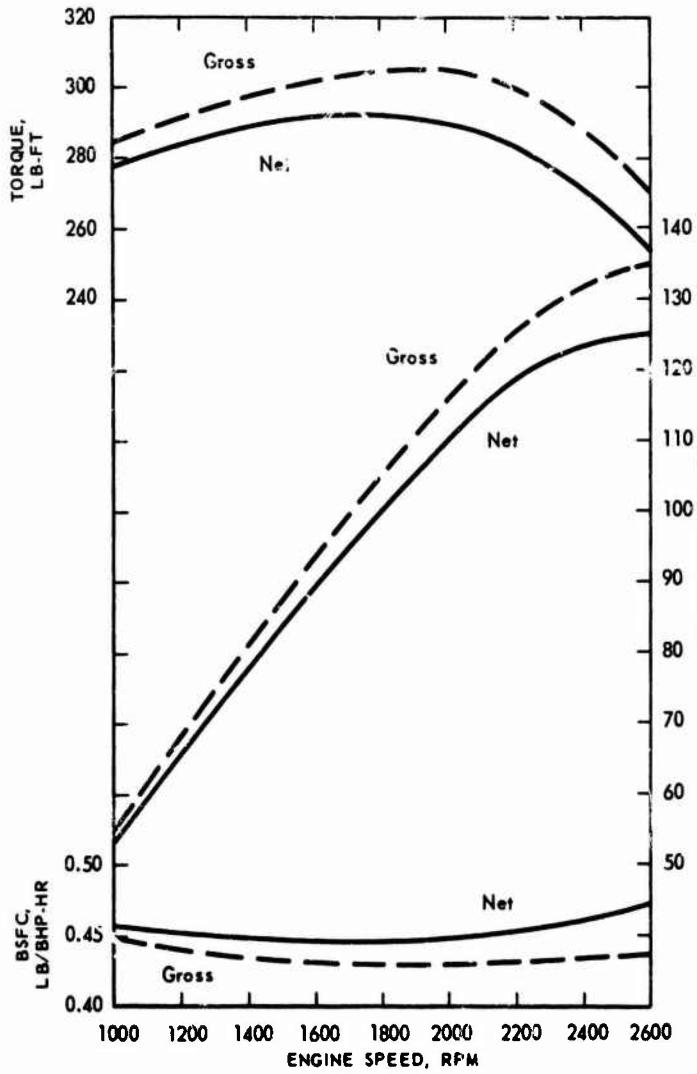


Fig. I-33—Performance Characteristics of AVM-310 Multifuel Engine with CITE Fuel

liquid-cooled engines with a cooling system. The AVM engine operates at an average brake mean effective pressure (bmep) of 60 psi at full speed-power. This is approximately half the bmep of conventional 4-cycle compression-ignition engines. This indicates that the bmep can be increased in the future as development progresses for higher power outputs. Turbosupercharging or supercharging can be incorporated for a further increase of approximately 25 to 30 percent in power output.

Fuel consumption for the AVM-310 series engine is shown in Fig. I-33. The fuel-rate curve shows a minimum or "best-point" gross brake specific fuel consumption (bsfc) of 0.425 lb/hp-hr and 0.435 lb/hp-hr at full power on CITE fuel. This fuel rate is slightly higher than that of either 4-cycle or 2-cycle uniflow scavenged engines in this power range, which operate at specific fuel rates of 0.39 to 0.41 lb/bhp-hr. However, the AVM fuel rate is considered good for a 2-cycle loop-scavenged engine in this power class. Further development work should reduce the fuel consumption to that approaching other compression-ignition engines.

Variable-Compression-Ratio Piston Engines

An R&D program was initiated in early 1960 to study the feasibility of increasing the power output of compression-ignition engines by the incorporation of a variable-compression-ratio (VCR) piston system coupled with an improved combustion-chamber design compatible with the VCR piston. The basic concept of the VCR piston system dates back to 1952, when the British Internal Combustion Engine Research Association (BICERA) initiated the design and development of a hydraulically actuated piston that varied compression ratio. The objective was to develop an automatic peak-pressure-control device that would increase power outputs without a corresponding increase in maximum combustion pressures. Peak pressure is one of the most critical limitations on diesel engine output. The subsequent successful development of a piston that varied the compression ratio from 15:1 to 8:1 allowed the output of the engine to be doubled without any increase in peak cylinder pressures.

Based on the successful development of the VCR piston system, the Continental Aviation and Engineering Corporation (CAE) obtained patent rights from BICERA for development and manufacturing rights in the US. Because of the obvious advantages of this system, ATAC has contracted with CAE to develop engines incorporating the VCR system for use in tactical vehicles.

The automatic hydraulically actuated VCR piston assembly operates somewhat like a hydraulic valve lifter and responds to peak cylinder pressures in much the same way that a hydraulic lifter responds to valve push-rod pressure. The VCR piston assembly consists of two main parts (see Fig. I-34): the piston shell or outer member A that carries the piston rings, and the piston-pin carrier or inner member B. The piston-pin carrier B, being linked to the crankshaft by the connecting rod and piston pin, always moves between fixed upper and lower limits, whereas the shell A is free to move within certain limits relative to the carrier B. The relative movement provides a variable height from the center of the piston pin to the top of the piston crown, thus effecting a variation in compression ratio through a change in the clearance volume without any change in engine displacement. The movement of the piston shell A is restrained hydraulically by the upper chamber C and an annular lower chamber D. The position of the two members A and B with respect to each other is determined by the control of the quantity of oil in the upper and lower chambers. Chambers C and D are filled with lubricating oil supplied through the nonreturn inlet valves H and J. Oil from the lubricating system is fed to the two valves via the passage E in the connecting rod (including a groove around both the wrist pin and connecting rod bearings), a spring-loaded slipper collector F, and the passages G in the carrier.

The upper chamber C is provided with a preset spring-loaded relief valve L. During compression and firing, this relief valve limits the maximum allowable pressure in chamber C and, in turn, the engine peak firing pressure. The relation between the pressures depends on the respective areas exposed to the two pressures.

The lower chamber D is provided with a fixed orifice K that controls the amount of movement between the two main members on the exhaust and intake stroke. Oil discharged from control orifice K and relief valve L returns directly to the crankcase. Note that as oil is bled from orifice K, decreasing volume D, volume C increases and is filled with oil through inlet valve H.

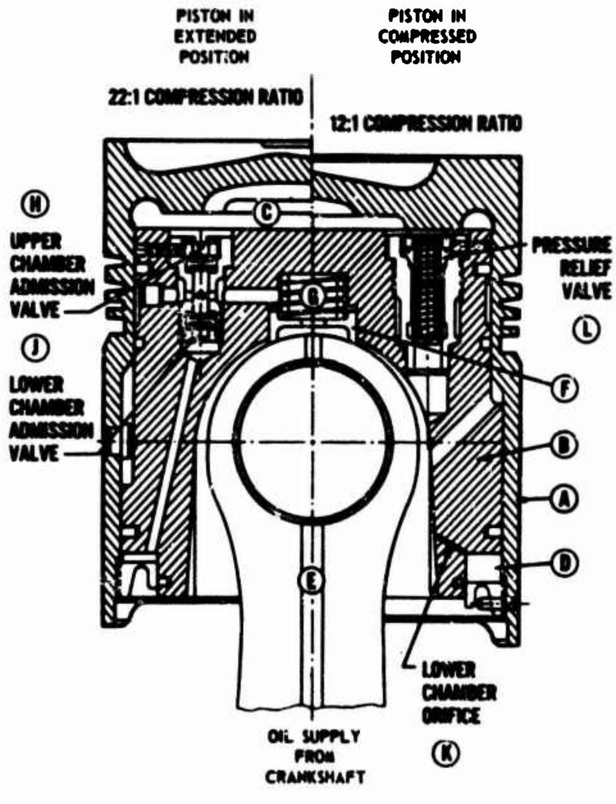


Fig. I-34—VCR Piston Assembly

The amount of oil discharged from the upper chamber in any single compression-power stroke depends on the margin by which the cylinder gas pressure exceeds that necessary to cause the discharge valve L to open and the duration of this excess pressure. The valve setting and the magnitude and duration of the excess gas pressure determine the rate of downward movement of the shell relative to the carrier. The upward relative movement, on the other hand, is the same on each exhaust induction stroke, being determined by the size of the fixed orifice K. If the upward and downward relative movements are equal, as is the case when the engine load pressure-time diagram remains constant, the effective compression ratio will remain unchanged. If the load is increased the downward movements will exceed the upward movements, lowering the compression ratio until an equilibrium is again established. Conversely, if the load is reduced, the compression ratio will be increased to a new point of equilibrium.

Changes in engine speed have virtually no effect on the amount of upward movement of the shell relative to the carrier, because the pressure in the lower chamber is created by an inertia force that varies as the square of the speed, while the rate of discharge from the fixed orifice varies with the square root of the pressure and thus directly with engine speed. Since cycle time is inversely proportional to speed the amount of oil discharged (rate × time) is independent of speed. This primary effect is only slightly modified by temperature variations and the viscosity of the oil.

During the initial assembly of an engine the relative position between the piston and the piston-pin carrier is not established. However, when the engine is operated initially and oil pressure is built up at the piston-pin bearing, the piston is automatically pumped up to the high limit of compression ratio, which facilitates starting. The piston shell remains at the extreme upper position as load is applied to the engine until the peak firing pressure exceeds the predetermined control pressure (about one-third to one-half full-load range). With a further increase in load the compression ratio adjusts automatically so that the peak firing pressure remains constant until the lower limit of compression ratio is reached. Once the lower limit is reached, the peak cylinder firing pressure will again increase, as illustrated in Fig. I-35. Conversely, as the load is reduced on the engine, the piston automatically increases the compression ratio to maintain the predetermined peak cylinder firing pressure. Thus when the engine is stopped, all pistons are at their maximum compression-ratio position.

The automatic operation of the VCR piston, as previously stated, is essentially the same as that of a hydraulic-valve-lifter tappet. The mechanism adjusts the height from the piston-pin center to the piston crown to a balanced condition and hence controls the compression ratio in accordance with the demands of the maximum allowable cylinder pressure.

The hydraulic operation of the VCR piston consists of the following sequence of events (see the schematic diagrams in Figs. I-34 and I-35). During the latter part of each upward stroke of the piston and the early part of each downward stroke, the inertia of the oil in the connecting rod, acting upward, creates a pressure in passages G. This pressure tends to open the inlet valves H and J and pump oil into the upper and lower chambers. At the same time the inertia of the piston shell, also acting upward, tends to raise the shell relative to the carrier. During the compression and power strokes the tendency is less than the tendency for opposite motion caused by the gas pressure on the piston crown, but not during the exhaust and induction strokes. Consequently during the latter part of each exhaust stroke and the early part of each induction stroke the shell moves upward relative to the carrier and oil enters upper chamber C through valve H. Simultaneously lower chamber D diminishes in volume, and oil is forced out of it via fixed orifice K. The lower chamber acts as a dashpot and restrains the tendency of the shell to seek the upper limit of its travel. Fixed orifice K is designed to ensure that the shell will not move upward relative to the carrier more than a small amount on each exhaust stroke.

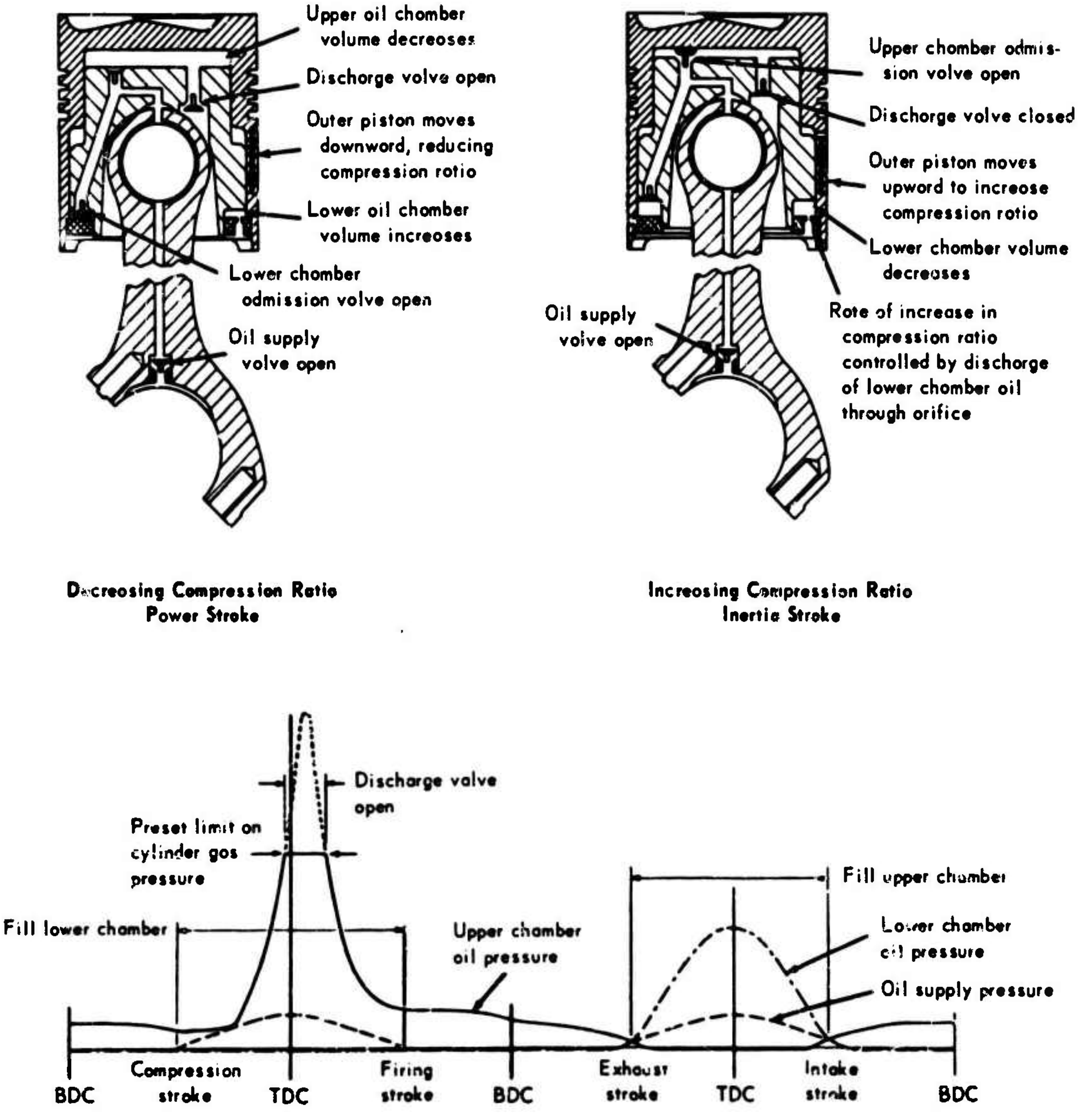


Fig. 1-35—Operation of VCR Piston System

bdc, bottom dead center; tdc, top dead center

During the compression and power strokes the gas pressure on the piston crown is transmitted to the carrier through the oil in the upper chamber, creating a high oil pressure in this chamber. Whenever the gas pressure exceeds the selected upper limit, sufficient oil pressure is built up in the upper chamber to open the discharge valve L and release some of the oil, allowing the shell to move downward relative to the carrier, thus decreasing the compression ratio of the engine. The downward movement of the shell enlarges the lower chamber D when the oil pressure in passage G is high; valve J opens as a result, and oil enters the lower chamber to keep it fully charged. There is no tendency for the piston to leak down while the engine is stopped since there is no pressure differential across any of the sealed joints.*

The illustration in Fig. I-36 shows an exploded view of the components that make up the VCR piston assembly.

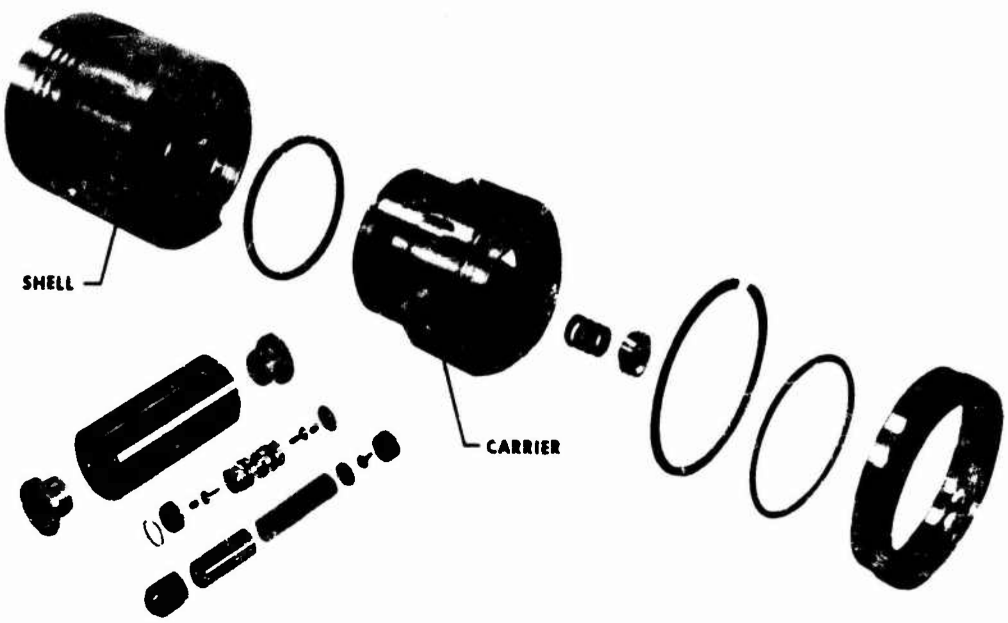


Fig. I-36—Exploded View of VCR Piston Assembly

The compression ratio of the VCR piston system can be varied to meet fuel requirements. Currently the VCR piston operates at a high compression ratio of 22 to 1 down to a low compression ratio of 12 to 1 or 10 to 1, depending on engine requirements. Figure I-37 illustrates a plot of compression temperature generated in the cylinder against compression ratio for various operating conditions from cold starting to full load. A plot of the required ignition temperature for gasoline and diesel fuel is superimposed on the graph. The curves are based on actual engine data and are from a compression-ignition engine having a bore of approximately 5 in., with direct fuel injection. The curve shows that a compression ratio of approximately 16 to 1 is required to

*This description of the operation of the VCR piston system is based on information furnished by the CAE.[3]

start the engine without starting aids when operating on diesel fuel, whereas at −25°F a compression ratio of 19 to 1 is required. The engine will idle satisfactorily with a compression ratio of 12 to 1 after starting. When gasoline is used as a fuel, starting aids are required even with a compression ratio of 24 to 1. However, the engine can be idled with a compression ratio of 18 to 1 and run at full power with a compression ratio of less than 12 to 1.

The major advantages of the VCR piston system are:

(a) Increases engine horsepower through peak-pressure control.
(b) Improves cold-weather starting characteristics.
(c) Enables the engine to operate with complete multifuel characteristics.

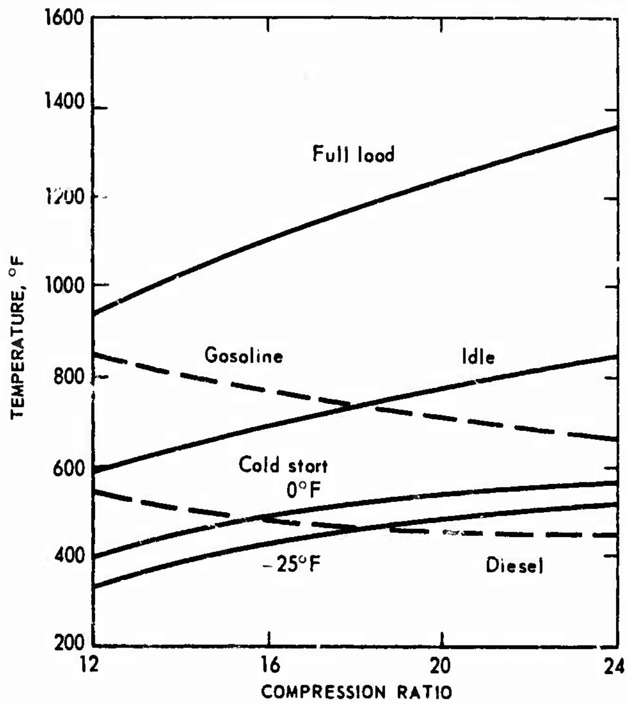


Fig. I-37—Ignition Characteristics as a Function of Compression Ratio for Diesel Fuel and Gasoline

——— Compression temperature
— — Ignition temperature

Several thousand hours on single-cylinder and multicylinder engines have been successfully completed to substantiate the theory and design of the VCR piston system. The VCR piston system was first applied to the conventional AVDS-1100 V-12 engine, which was originally rated at 550 bhp at 2800 rpm. With the VCR piston system this engine developed 825 bhp without any change in engine structure. This represents an increase of 50 percent in horsepower. Further development and testing have increased this output over 100 percent above that of the engine in its conventional configuration.

The VCR piston system can be applied to any 4-stroke-cycle engine, and possibly to 2-stroke-cycle engines as well.

Discussion. The VCR piston system can be applied to any cylinder arrangement and configuration, such as Vee, opposed, or in-line. The family capabilities are unlimited depending on bore size, the smaller-bore cylinders having better family capabilities than the large-bore sizes.

Engines incorporating the VCR piston system can be either liquid-cooled or air-cooled with equal success. The VCR piston engines can be manufactured using present automotive-engine tooling, since the power-producing components are similar to the power-producing components of more conventional reciprocating-piston compression-ignition engines.

As a result of the successful research, development, and test work that is being carried on, the present technical status and projected advancement levels of the VCR engine program can be summarized thus: Basic design parameters and limitations, such as bore, stroke, bmep, rpm, turbosupercharging, efficiencies, flow characteristics, pressure ratios, compression ratios, combustion limits, heat transfer and heat-load limits, injection limits, piston design and control, bearing capacities, structural limits, etc., have been established to verify that the VCR piston-engine concept at a power output of 250 bmep at 2800 rpm is a practical development today. A VCR engine system with an output of 260 to 280 bmep could be developed with a good probability of success.

Present applications of the VCR piston system that are under development are to the liquid-cooled hypercycle engine and to the US/FRG (Federal Republic of Germany) main battle tank engine. The hypercycle engines (the naturally aspirated 120 bhp LD-465-1 and the turbosupercharged 210 bhp LDG-465-1) have been adapted to incorporate the VCR piston. This new engine, designated the LVCR-465 by the commercial sponsor (CAE), is shown in Figs. I-38 and I-39. The LVCR-465 specifications are shown in Table I-5. The hypercycle 465 $in.^3$ displacement engine is rated at 140 bhp naturally aspirated and 210 bhp turbosupercharged. By application of the VCR piston the rating has been increased to 350 to 450 bhp. This represents a power increase of several times that of the basic engine. Figure I-40 illustrates a performance comparison between the naturally aspirated LD-465-1 engine, the turbosupercharged LDS-465-1 engine, and the same engine with VCR pistons—the LVCR-465 engine. The horsepower curve indicates a 220 percent increase with VCR pistons and turbosupercharging over the basic 140-bhp engine.

The AVCR-1100-2 engine is currently being developed for the US/FRG main battle tank. This engine is the outcome of several lower-powered engines that were abandoned when the power requirements for the main battle tank were increased. The AVCR-1100-2, illustrated in Fig. I-41, is a 4-stroke-cycle air-cooled V-12-cylinder compression-ignition engine incorporating the VCR piston system.

The engine is constructed of aluminum for minimum possible weight while retaining heavy-duty structural integrity through the use of advanced design techniques. The engine is designed to operate at much higher temperatures by the incorporation of a unisteel cylinder-head assembly. It features a finely spaced deep-finned aluminum outer head assembly, which is aluminum-bonded to a one-piece steel inner head assembly, thus providing a one-piece head as-

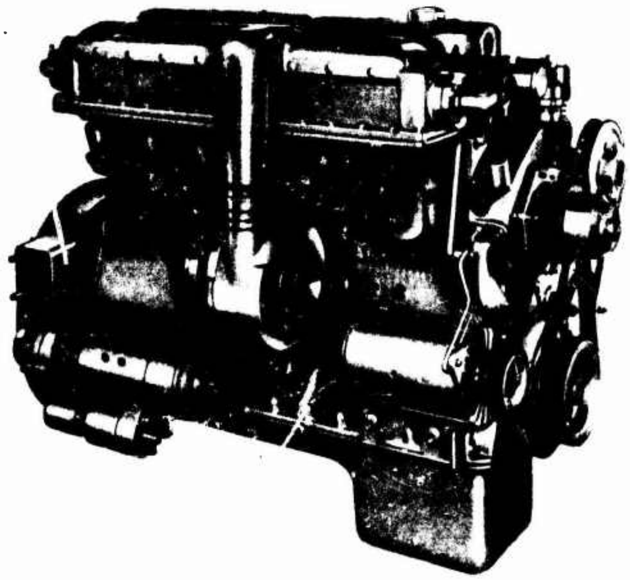


Fig. I-38—Right Front View of Model LVCR-465 Multifuel Engine

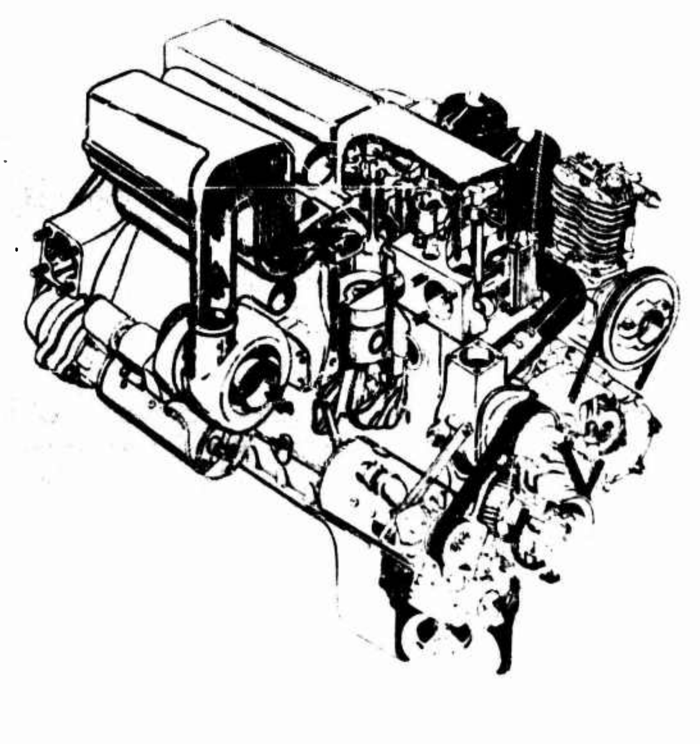


Fig. I-39—Cutaway of LVCR-465 Engine

TABLE I-5

Principal Specifications of the LVCR-465 Engine

Item	Value or description
Model	LVCR 465
Type	6-cylinder, in-line, liquid-cooled, multifuel compression ignition engine with variable-compression-ratio pistons
Induction system	Supercharged by an exhaust-driven wastegate turbocharger
Bore and stroke, in.	4.56 × 4.88
Displacement, in.^{3}	478
Compression ratio	22.0:1 (high) to 12.0:1 (low)
Horsepower, gross, at rated rpm	450 hp at 2800 rpm; 266 bmep (psi) 400 hp at 2800 rpm; 236 bmep (psi) 350 hp at 2800 rpm; 206 bmep (psi)
Idle speed, rpm	600 to 700
Maximum torque, lb-ft	
At 450-hp rating	900 at 2000 rpm
At 400-hp rating	800 at 2000 rpm
At 350-hp rating	750 at 2000 rpm
Valves	Overhead type, two per cylinder, actuated by a single camshaft
Cooling system	Thermostatically controlled, pump-circulated, liquid-cooled, with a balanced rotor manifold. Solid or viscous-clutch mounted fan for radiator air circulation
Overall dimensions (complete engine)	
L × W × H, in.	44.0 × 28.9 × 39.5
Volume, ft^3	29
Engine weight (complete engine, dry), lb	Cast iron, 1725; aluminum, 1355
Specific power output (at 450 bhp rating), bhp/ft^3	15.5
Specific weight (at 450 bhp rating), lb/bhp	Cast iron, 3.8; aluminum, 3.0

sembly. This single-piece unit-structure cylinder head was developed to provide the structural strength and the high temperature capabilities necessary to support the high horsepower output produced by the VCR piston. The VCR piston itself provides built-in piston-oil cooling, thus providing the engine with higher output capabilities. An open-modified-swirl combustion-chamber design is used in the engine, and fuel is sprayed directly into the combustion chamber through a multiple-orifice fuel-injection nozzle.

The specifications of the AVCR-1100-2 engine are shown in Table I-6. The engine is tentatively rated at 1200 bhp (100 bhp/cylinder) owing to the unavailability of adequate turbosupercharger and fuel-injection units. Fabrication of the 1475-bhp (123 bhp/cylinder) unit is anticipated to begin late in 1966. Indications are that the AVCR-1100 engine will ultimately be limited in top power output at approximately 1700 to 1800 hp owing to structural limitations of the engine. It is also believed that oil will be a limiting factor due to coking

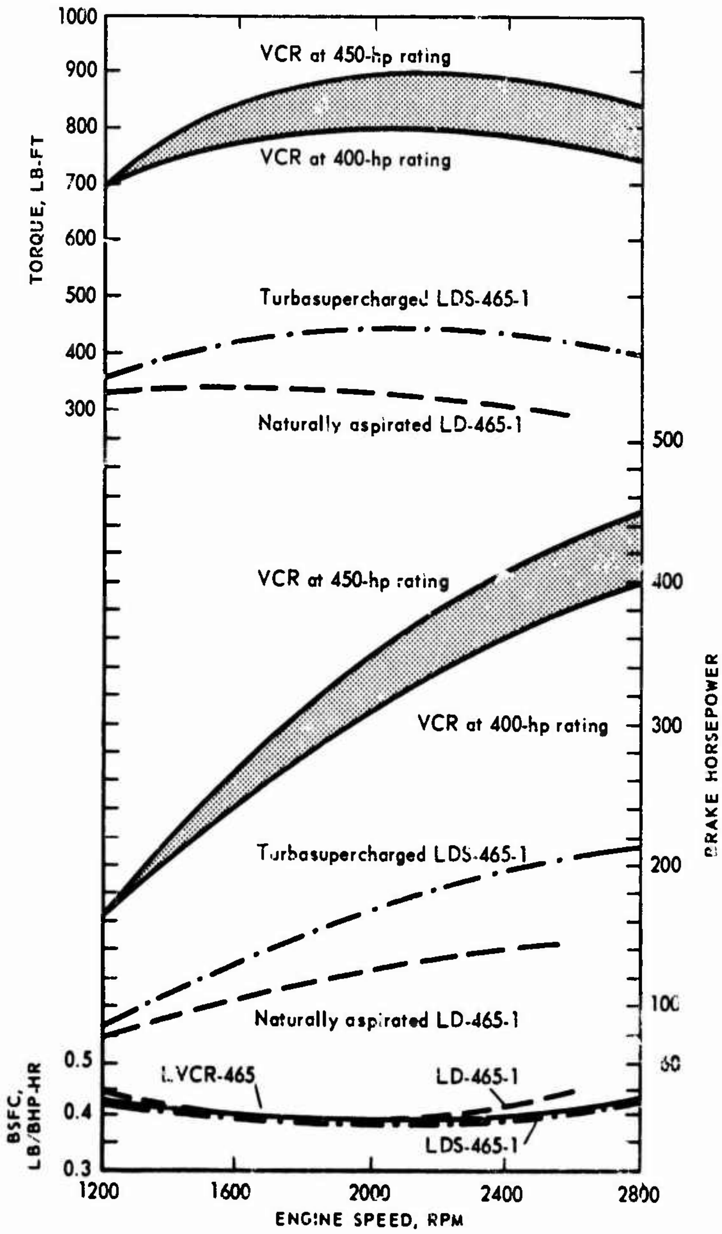


Fig. I-40—Comparison of Performance Characteristics of 465-in.^{3} Displacement Engines: Naturally Aspirated, Turbasupercharged, and with a VCR Piston System

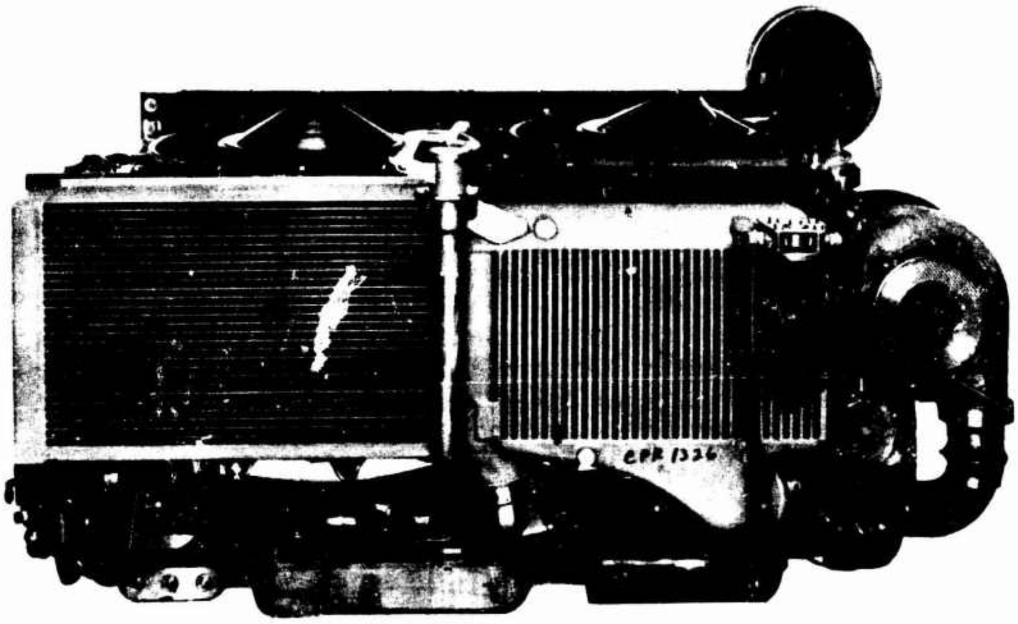


Fig. I-41—1475-hp AVCR-1100-2 Engine

TABLE I-6

Principal Specifications of the AVCR-1100-2 Engine

Item	Value or description
Configuration	120-deg V-12
Displacement, $in.^3$	1120
Bore and stroke, in.	4.875 × 5
Compression ratio	Variable: 22:1 to 10:1
Gross horsepower	1475; 370 bmep (psi)
Rpm rating	2800
Minimum fuel consumption, lb bhp-hr	0.39
Gross maximum torque, lb-ft	2750 at 2200 rpm
Cooling	Air cooled
Cooling-fan power,[a] hp	172
Cooling air flow, cfm	31,000 at 60°F
Induction air flow, cfm	3,500 at 60°F
Overall dimensions (complete engine) L × W × H, in.	69.0 × 60.6 × 35.0
Envelope volume, ft^3	85
Bare-engine envelope volume, ft^3	~ 58
Bare-engine weight,[b] lb	3,200
Installed weight, dry, lb	3,740
Specific power output, bhp ft^3	
Complete engine	17.35
Bare engine	25.5
Specific weight, lb bhp	
Complete engine	2.56
Bare engine	2.17

[a]Includes all cylinder, engine oil, aftercooling, and transmission (11,000 btu min) oil cooling.

[b]Bare engine is less coolers, fans, or starter.

between the inner and outer piston shell caused by the severe pressures and temperatures generated in this area. However, synthetic oil such as that used in turbine engines could eliminate coking if it becomes a problem.

The power increase due to the VCR piston has a marked effect on the specific engine weight and volume. The liquid-cooled VCR engines, such as the 400- to 450-hp LVCR-465, have a specific weight of 3.0 to 3.4 lb/bhp and a specific power output of 15.5 bhp/ft^3 of volume. When specific outputs of a liquid-cooled engine are compared with those of an air-cooled engine, the weight and volume of the cooling system must be considered for a true comparison. It generally follows that for military power plants the air-cooled engines, complete with integral cooling systems, are somewhat lighter and less bulky than liquid-cooled engines when compared as a completely installed power package.

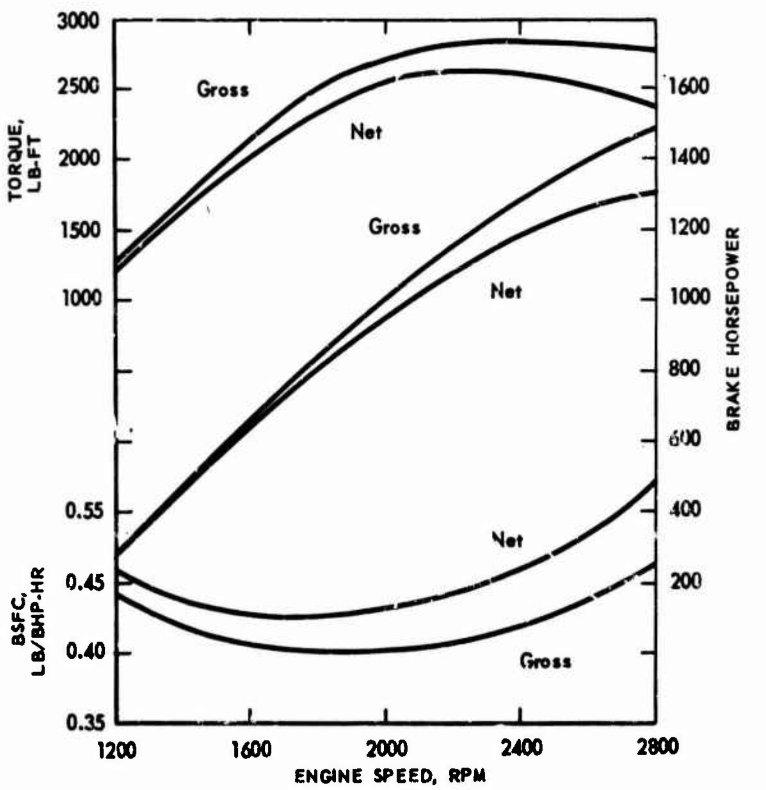


Fig. I-42—Estimated Full Rack Performance of AVCR-1100 Engine with CITE Fuel

The air-cooled 1475-hp AVCR-1100 engine has a specific weight of 2.56 lb/bhp and a specific-power output of 17.35 bhp/ft^3 for the complete unit-cooled engine assembly. When the "base" engine minus the cooling-system components is considered for comparison with a liquid-cooled engine, the specific weight is 2.17 lb/bhp and the specific power output is 25.5 bhp/ft^3 of volume.

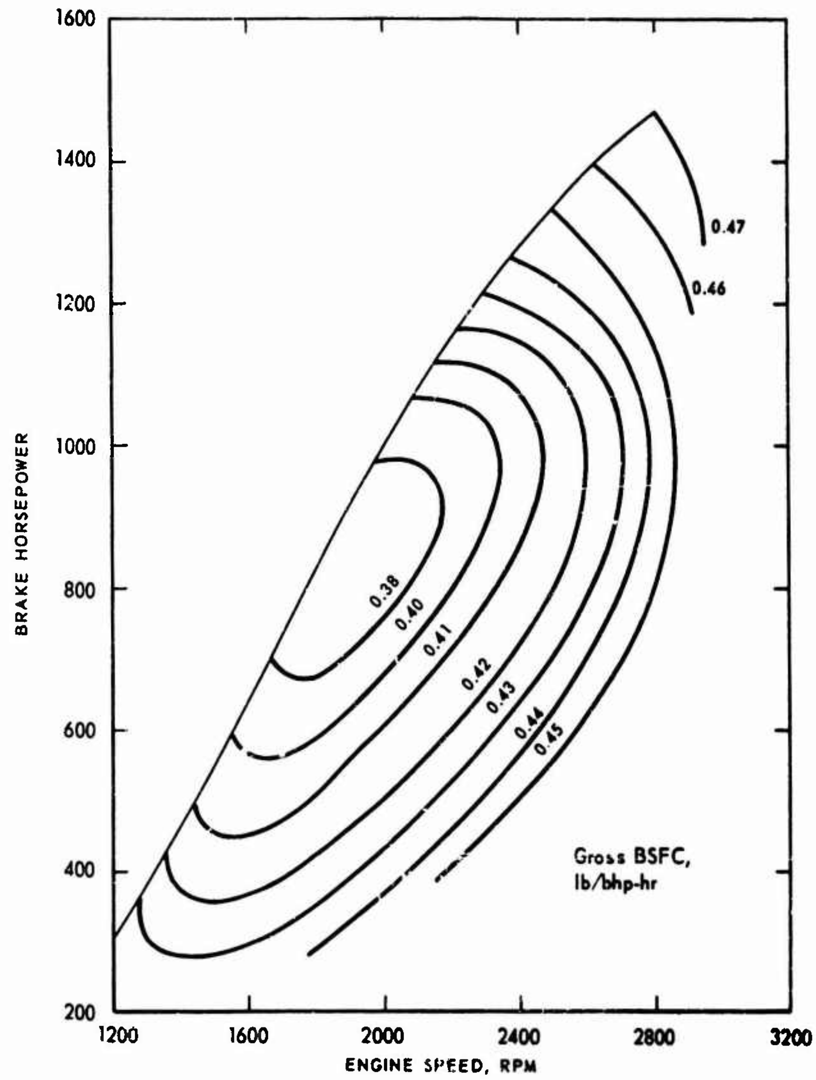


Fig. I-43—Fuel Map for 1475-bhp AVCR-1100-2 Engine
CIE fuel, series supercharging

The fuel-consumption performance of engines incorporating the VCR piston is comparable to that of high-performance commercial compression-ignition engines. As shown in Fig. I-40, the best-point fuel rate for the LVCR-465 engine is 0.38 lb/bhp-hr at part load and 0.44 to 0.45 lb/bhp-hr at full power. The performance characteristics of the AVCR-1100-2 engine are shown in Figs. I-42 and I-43. The fuel-rate map (Fig. I-43) indicates a minimum part-load fuel rate of 0.36 lb/bhp-hr and 0.46 to 0.47 lb/bhp-hr at full power. Engines incorporating the VCR piston system have demonstrated good multi-fuel performance capabilities.

The combination of an advanced combustion system, such as the very-high-output (VHO) system, with the VCR piston system can eventually further reduce fuel consumption and increase power output.

Very-High-Output Engines

An R&D program was initiated in 1960 to provide the armed services with a lightweight, compact, reliable, very-high-output multifuel power source for military vehicles. To accomplish the above objectives it was necessary to advance the present state of the art by refining and improving existing systems and components. A new fuel-injection valve, precombustion chamber, and main combustion chamber design were developed to improve the combustion process necessary to achieve the desired multifuel characteristics, low fuel consumption, and high power output. The development of the VHO engine was contracted to the Caterpillar Tractor Company.

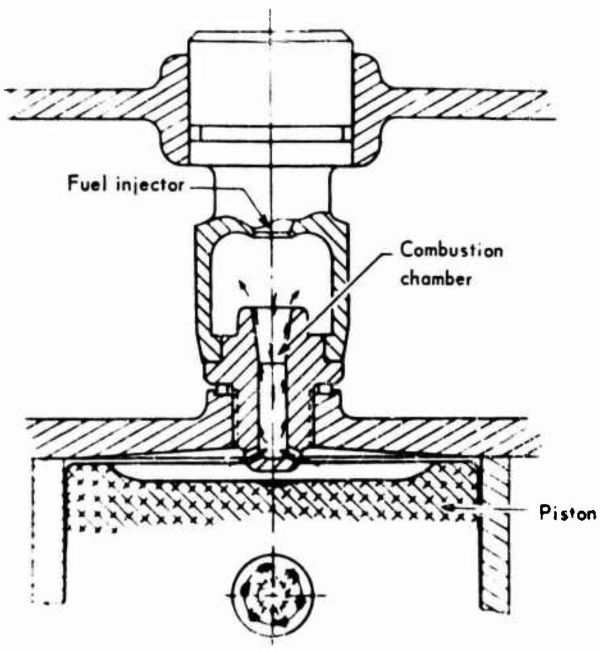


Fig. I-44—VHO Combustion System

The VHO engine is a turbosupercharged 4-cycle, uniflow compression-ignition engine. The research program objectives were to develop a single-cylinder test engine that would be capable of producing acceptable performance outputs up to 400 psi bmep at engine speeds of 2800 rpm. The data obtained could then be used to determine whether these high outputs could result in a compact, lightweight multifuel multicylinder engine. Several thousand hours of operation on single-cylinder and multicylinder test engines were successfully completed to substantiate the theory and design of the engine and its component systems. The VHO engine employs high-pressure-ratio turbosupercharging with air-to-air combustion-air aftercooling and a precup turbulent-chamber combustion system. A schematic diagram of this system is shown in Fig. I-44.

Fuel is injected through a fuel-injection valve into the precombustion chamber. Since the amount of air in the chamber is limited, only part of the fuel burns. The resulting heat and pressure expel the remaining fuel-air mixture into the main combustion chamber. An air swirl is created by holes located at the entrance of the prechamber. The air is then directed up the side of the throat of the chamber, creating a partial vacuum in the center of the throat into which fuel is sprayed. The result of injecting fuel into the vacuum is that only the fine fringes of the fuel spray are stripped off by the inrushing air. The fine fuel droplets in the fringe quickly vaporize and ignite, serving to trigger the burning of the remainder of the fuel charge for smooth, even combustion. The basic concept of the VHO combustion system has evolved from a commercial engine of similar type, size, and configuration that has a proved history of excellent durability and reliability at relatively high power output.

The VHO engine program has benefited greatly from the Army development of the LVDS-1100 and LDS-750 series engine programs, which became outdated during development and were subsequently terminated owing to a requirement for greater horsepower for tank propulsion. However, these engines had produced power outputs approximately 15 percent over their design goals. Much design knowledge and experience were gained from this program on the use of aluminum material, structural design, bearing capacities, fuel sensitivity, and aftercooling all of which have been very useful to the VHO and other engine programs.

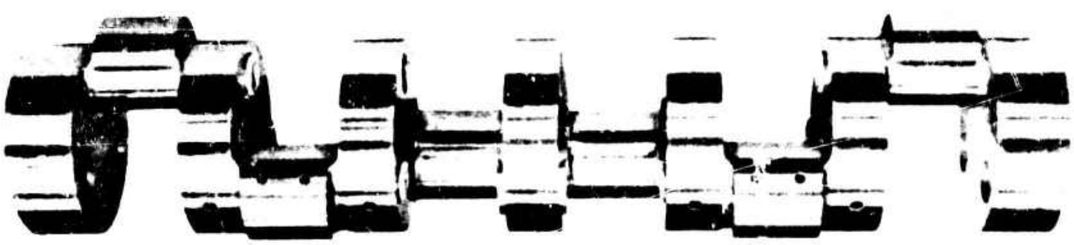


Fig. I-45—Unique Barrel-Type Crankshaft for VHO Engine

The VHO engine is constructed of aluminum to achieve low weight while retaining structural integrity for heavy-duty operations. The engine features a four-valve head for increased volumetric efficiency and a unique barrel-type crankshaft, shown in Fig. I-45, which eliminates separate crank-cheeks and results in a compact design. The large circular disks serve as both main bearing journals and crank throws. The VHO engine is designed to operate at high coolant temperatures, thus requiring a smaller and more efficient cooling system. A unique design feature of this power plant is the provision for alternative locations for such components as the turbosuperchargers, induction and intake, hydraulic pump, water pump, generator, starter, and other accessories, without modification of the basic engine. This feature provides the vehicle designer with the flexibility to select the optimum engine configuration for a given vehicle application. Based on the outcome of performance and durability testing, a 12-cylinder engine design has been completed and fabricated, and is currently undergoing testing and further development.

Discussion. The VHO engine is of Vee-type configuration. The present full-sized development engine, shown in Fig. I-46, is a 12-cylinder unit that develops 1000 bhp. It is one of a possible family of engines that can be built around the same bore size by varying the number of cylinders. Because of its relatively small (4.5 in.) bore size, the engine design lends itself particularly well to the family concept. The family capabilities of the VHO engine are shown in Table I-7.

The VHO engines are designed around liquid-cooling principles. However, air-cooling of these units should be as practical as with other air-cooled compression-ignition engines. A maximum commonality of parts and components characterizes all engines of the family. These engines can be manufactured using present automotive-engine tooling since the power-producing components are similar to the reciprocating components of more conventional compression-ignition engines.

Basic design parameters and limitations, such as bore, stroke, bmep, rpm, turbosupercharging, efficiencies, flow characteristics, pressure ratios, intercooling temperatures and capacity, combustion limits, heat-transfer and heat-load limits, injection limits, bearing capacities, structural limits, etc., have been established to verify that the VHO engine concept, at an output of 240 to 260 bmep and at 2800 rpm, can be developed with a high probability of success. Indications are that with continued development effort the VHO engine system can produce outputs of 260 to 300 bmep at 2800+ rpm with a good probability of success.

The 12-cylinder VHO engine has a specific weight of 2.5 lb/bhp and a specific power output of 31 bhp/ft^3 of volume. The smaller engines of the proposed family (see Table I-7) have specific weights of 3.8 to 2.9 lb/bhp and specific power outputs of 19 to 27 bhp/ft^3 of volume.

The fuel consumption of the VHO engine is 0.38 lb/bhp-hr minimum or best-point operation. This fuel rate compares with conventional high-performance commercial compression-ignition engines. The peak cylinder pressures obtained when using gasoline are about the same as when diesel fuel is used.[4] The power of this engine when running on gasoline was 88.2 percent of its power with CIE fuel.[5] The engine has demonstrated multifuel capabilities and can operate satisfactorily on a fuel range from military gasoline to diesel fuel, as well as some crudes.[6]

The VHO combustion principle can be applied to other engines, both large and small. It appears likely that the VHO system can result in much greater power outputs when applied to the VCR piston concept, which was discussed previously in this chapter.

Extremely-High-Output (Kamm) Engine

The development of very-high-pressure compression-ignition engines was investigated in Germany in the late thirties and early forties, where test engines were constructed and operated. Although these early units were plagued with many problems, primarily involving metals, the results appeared very promising. In the early fifties several US organizations continued the investigation of the high-pressure compression-ignition engine.

The US Army recently initiated an R&D program with the Stevens Institute of Technology to determine the feasibility and practicality of obtaining extremely

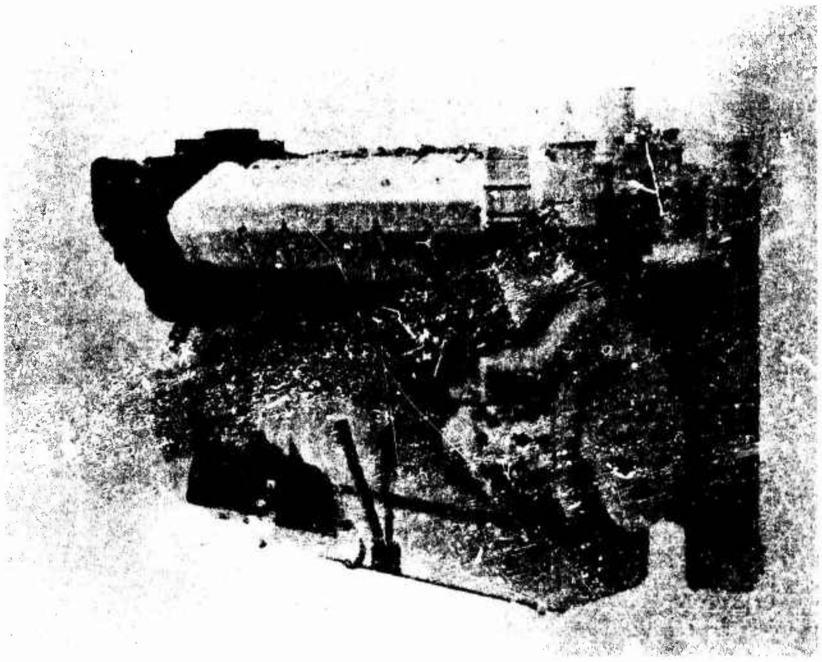


Fig. 1-46—Three-Quarters-Front View of 12-Cylinder 1000-hp VHO Engine

TABLE 1-7

Characteristics of VHO Engine

Item	Model			
	LMS-350	LMS-525	LVMS-700	LVMS-1050
Type and no. of cylinders	In-line, 4	In-line, 6	60-deg V-8	60-deg V-12
Bore and stroke, in.	4.5 × 5.5	4.5 × 5.5	4.5 × 5.5	4.5 × 5.5
Displacement, in.^{3}	350	525	700	1050
Horsepower, max, at 2800 rpm	330	500	665	1000
Minimum fuel consumption, lb/bhp-hr	.38	.38	.38	.38
Length, in.	35.2	47.0	36.9	48.6
Width, in.	24.6	24.6	32.1	32.1
Height, in.	35.6	35.6	35.9	35.9
Volume, ft^3	17.4	23.3	24.5	32.5
Dry weight, lb (less fan, radiator, generator)	1250	1625	1900	2500
Specific output, bhp/ft^3	19.0	21.5	27.0	31.0
Specific weight, lb/bhp	3.8	3.3	2.9	2.5

high power output from a lightweight compact reciprocating compression-ignition engine.

The extremely-high-output (EHO) engine is a highly turbosupercharged 2-cycle loop-scavenged compression-ignition engine with a combustion system that provides precise combustion control. Optimum combustion enables operation at very high bmeps. A schematic diagram of this system is shown in Fig.

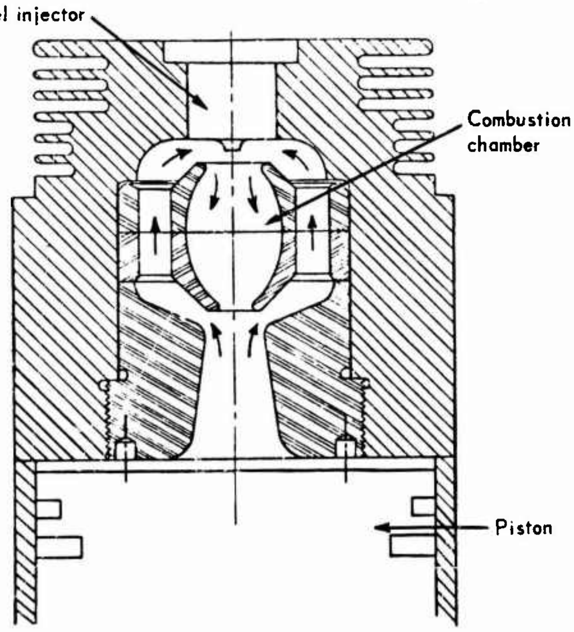


Fig. I-47—EHO Combustion System

I-47, and a cross section of the cylinder assembly of the experimental test engine is shown in Fig. I-48. This control is accomplished by keeping air and fuel separated in the combustion chamber except for a small portion of air that is allowed to mix with fuel at a controlled rate to enter into combustion. This system is capable of operating at higher thermal loads by confinement of high temperatures to a small area in the combustion chamber. The piston design, shown in Fig. I-49, insulates critical parts from high temperatures. The EHO combustion system has demonstrated a high degree of insensitivity to fuel characteristics. Experimental work is continuing to establish the limitations and capabilities of the system and to obtain better understanding of the concepts and principles on which the engine design is based.

Discussion. The EHO engine is in the research stage, and only experimental test engines have been produced. If successfully developed, the EHO engine can be constructed in either Vee, in-line, or opposed-cylinder configurations and would incorporate family capabilities similar to those of a conventional piston engine. The present single-cylinder test engine is air-cooled but either air cooling or liquid cooling of these engines is practical. The EHO engine would be more difficult to produce than present compression-ignition engines owing to the greater complexity of its combustion chamber and cylinder

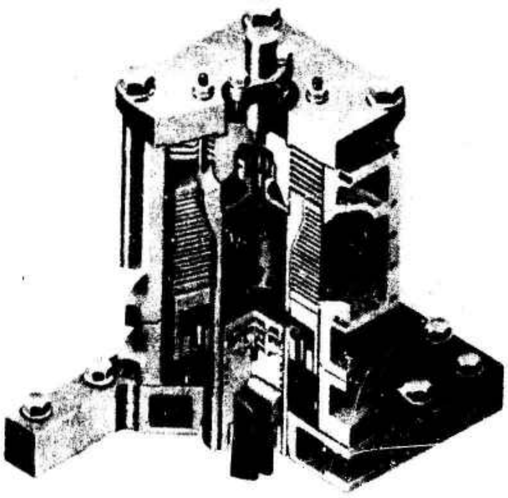


Fig. I-48—Cylinder Cross Section of EHO Experimentol Engine

Fig. I-49—Cross Section of EHO Piston Assembly, Showing Heot-Protective Crown with Fire Rings and piston assemblies. However, the engines can be manufactured using most of the present automotive-engine tooling since the power-producing components are similar to the components of conventional engines.

Test results have been very encouraging; outputs of 350 psi bmep[7] (equivalent to an output of 700 psi in a 4-cycle engine) have been achieved. A significant accomplishment has been the operation of this engine at a bmep of 360 psi, which corresponds to 450 psi imep (indicated mean effective pressure) for ½ hr under steady-state conditions. However, emphasis has been shifted from high performance to durability operation at a bmep of 250 psi. Well over 1500 hr of operation have been accumulated with bmep outputs to 250 psi and well over 200 hr at bmep outputs of from 250 to 300 psi.

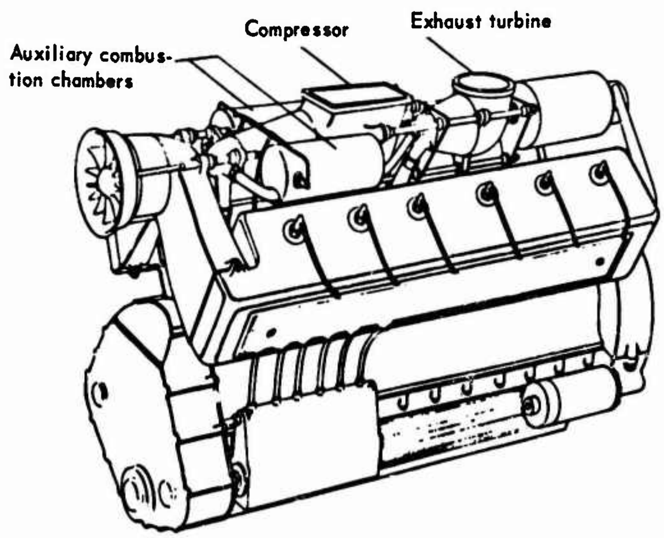


Fig. I-50—Proposed EHO 12-Cylinder Engine

It is estimated that a 12-cylinder EHO engine, with a 2.75 in. bore and a 4.5 in. stroke, operating at a bmep of 250 psi at 3000 rpm would produce 590 hp for a specific output of 29 hp/ft^3 and a specific weight of 2.02 lb/hp. Higher bmep and torque over a wide speed range can be achieved by increasing the boost pressure independent of engine speed. One method of achieving this is by incorporating an exhaust-driven gas turbine, whereby the reciprocating engine delivers all the torque and the mechanically independent turbocharger supplies the necessary boost. Additional gas energy can be supplied to the turbine section by means of an external bypass combustion chamber located between a compressor and the gas turbine, as shown in Fig. I-50. This section will supply additional gas energy, which cannot be supplied by the energy available in the compression-ignition engine exhaust, under certain power-demand conditions. However, this fuel could not be used as efficiently as it would be if it were added to the diesel section of the engine. Consequently fuel consumption would be very high when this section is operating. (A similar system is discussed in the section on compound engines.) It is, however, a means of attaining high power outputs for short periods of time. Another method would be to utilize a differentially driven Rootes-type supercharger to supply the extra boost. This system would provide better utilization of fuel energy but probably would not yield the high power outputs that the burner-augmented system would provide. The weight of the total engine package would also increase appreciably.

The specific fuel consumption of test engines is between 0.30 and 0.40 lb/indicated horsepower (ihp)-hr at outputs up to 250 psi bmep and 0.50 lb/ihp-hr at top output of 450 psi imep. These are indicated fuel rates, and the bsfc would actually be about 20 percent higher.

Estimated performance and physical characteristics of possible 6-, 8-, and 12-cylinder EHO engines are shown in Table I-8.

TABLE I-8

Estimated Characteristics of EHO Engines

Item	Model					
Type and no. of cylinders	V-6		V-8		V-12	
Bore and stroke, in.	2.75 × 4.5		2.75 × 4.5		2.75 × 4.5	
Displacement, in.^{3}	156		208		312	
	Cruise	Maximum	Cruise	Maximum	Cruise	Maximum
Horsepower:						
At 2400 rpm	210	310	300	410	460	660
At 3000 rpm	270	390	380	540	590	830
Bmep, psi	250	350	250	350	250	350
Torque, ft-lb at 2400 rpm	460	680	655	920	1035	1450
Length, in.	30		36		46	
Width, in.	25		25		25	
Height, in.	30		30		30	
Volume, ft^3	13		16		20	
Weight (dry), lb	680		920		1200	
Specific weight, lb/hp:						
Cruise hp at 2400 rpm	3.24		3.07		2.61	
Cruise hp at 3000 rpm	2.52		2.42		2.02	
Max. hp at 2400 rpm	2.20		2.19		1.82	
Max. hp at 3000 rpm	1.75		1.70		1.45	
Specific output, hp/ft^3:						
Cruise hp at 2400 rpm	16		19		23	
Cruise hp at 3000 rpm	21		24		29	
Max. hp at 2400 rpm	24		26		33	
Max. hp at 3000 rpm	30		34		41	

HYBRID ENGINES

In the past few years considerable research work has been carried on by the military, research organizations, major oil companies, and other private commercial concerns to develop an advanced high-efficiency high-output combustion process for application to both spark- and compression-ignition engines. The primary objectives of these programs are to develop an ideal combustion process that would yield the maximum possible output and performance characteristics per cubic inch of engine displacement with a very high degree of fuel economy from varied low-grade fuels—all contained within a lightweight, compact package.

To accomplish these objectives a great deal of R&D effort by both industry and the military has been expended on the hybrid combustion system, or "hybrid engine." The hybrid engine is simply defined as "an engine that is a cross between a spark-ignition engine and a compression-ignition engine." The hybrid engine operates on a lean fuel mixture to eliminate throttling losses, like a compression-ignition engine, and uses electrical energy to initiate combustion, like a spark-ignition engine. This combustion system enables the hybrid engine to approach the fuel economy of the compression-ignition engine. The hybrid system offers increased fuel economy while retaining the weight, size, structure, and cold-starting ability of the spark-ignition engine.

There are many different types of hybrid combustion systems. Some operate almost as compression-ignition engines, and some operate almost as pure spark-ignition engines with external fuel mix. The conventional spark-ignition engine functions with external fuel and air mixing and directs a homogeneous charge into the combustion chamber. A fixed-point electric spark is used to initiate combustion, which then propagates a flame front throughout the fuel-air mix in the chamber. Air and fuel regulation (throttling) is necessary to control engine speed and power output within the narrow range of ignitable air-fuel ratios. The conventional compression-ignition engine functions with internal fuel and air mixing, which creates a heterogeneous mixture within the combustion chamber, thus providing a variety of air-fuel ratios. Combustion of the fuel is by the heat of compression and is uniform through the combustion chamber because a variety of localized air-fuel ratios ensures initiation of burning. A wide range of fuel distillates can be ignited in this manner. Only the fuel is regulated to control engine speed and power output, eliminating the losses associated with the throttling of air. The major difference between the Otto cycle and the diesel cycle, apart from their ignition principles, is their air-fuel ratio operating range. The diesel engine cannot be operated richer than stoichiometric, and the spark-ignition gasoline engine cannot be operated much leaner than stoichiometric, both being controlled by smoke limits.

The majority of the well-known hybrid combustion systems operate on the so called "stratified-charge" concept, where a small amount of the total injected fuel is directed in a rich mixture near the point of ignition. The major portion of the fuel charge in an excessively lean mixture is injected toward the periphery of the combustion chamber. The small rich mixture is electrically ignited and propagates a flame to ignite the lean fuel mixture.

Hybrid engines operate on a stoichiometric air-fuel mixture at full-load operation and on excessively lean (60 to 1 and higher) air-fuel mixtures at part-load operation. High thermal efficiency and fuel economy are attained by those engines through their ability to burn lean air-fuel mixtures and a wide range of fuels that would not require any specific octane and cetane rating. Because the maximum cylinder combustion pressures as well as the rate of pressure rise is less in the hybrid combustion process than in the diesel cycle, engine construction could be considerably lighter. The hybrid engine could be built around the conventional spark-ignition engine structure, resulting in a lightweight compact low-cost engine for military vehicles.

Ricardo Stratified-Charge System

The Ricardo stratified-charge system, shown in Fig. I-51, was developed in the 1920's. This approach to stratification consisted of a precombustion chamber (or ignition cell) with a spark plug located in the cell. A rich mixture was ignited in the cell by the spark plug. The burned mixture then rushed through a narrow neck into the main combustion chamber to ignite a leaner mixture. Good power output and fuel economy were achieved.

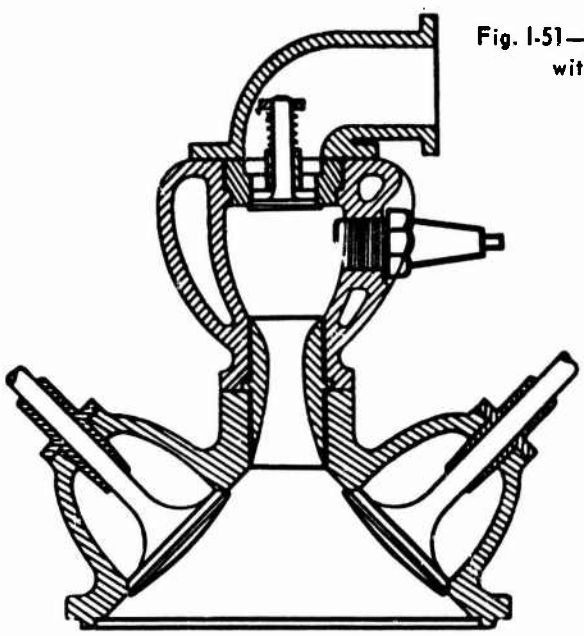


Fig. I-51—Ricordo Strotified-Chorge System with Precombustion Chomber

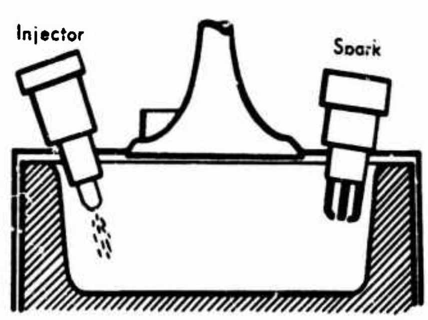


Fig. I-52—Hesselmon Combustion System

Hesselman System

The Hesselman "low-pressure diesel engine" was an attempt to combine the Otto and diesel cycles. This system, shown in Fig. I-52, utilized a shrouded intake valve to induce an air swirl. A fuel nozzle at the periphery of the combustion chamber, which was located in the top of the piston, injected fuel toward the center of the combustion chamber during the compression stroke. A spark plug located diametrically opposite the fuel nozzle ignited the fuel charge. The engine operated at a compression ratio of 8 to 1. The air was throttled at idle and low speeds only, with fuel input controlling the higher speed-power loads.

Dysart-Freeman System

The Dysart-Freeman combustion system, shown in Fig. I-53, is a stratified-charge concept that utilizes a prechamber containing a spark plug and fuel-injection nozzle. The ignited charge is rushed from the prechamber into the main combustion chamber to ignite the main mixture supplied by a fuel-injection nozzle.

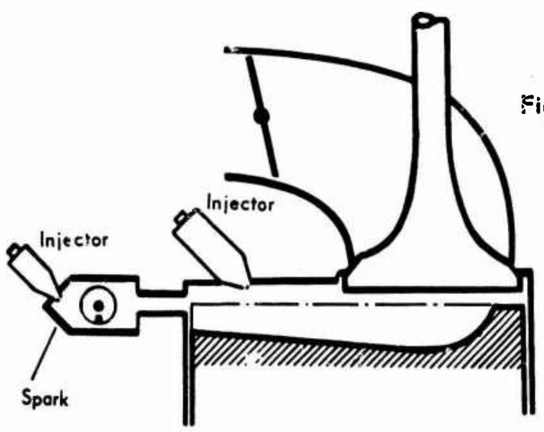


Fig. I-53—Dysart-Freeman Combustion System

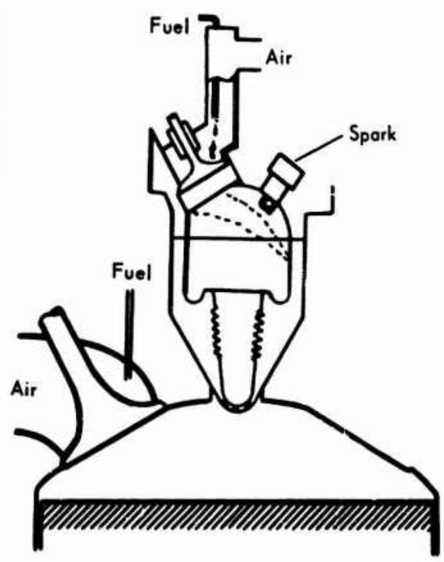


Fig. I-54—Heintz Ram-Straticharge System

Baudry-Stuempfig System

The Baudry-Stuempfig system utilizes a small bent tube that carries a rich fuel mixture close to the intake valve. The tube directs the rich mixture toward a spark plug located at the periphery of the combustion chamber.

Heintz Ram-Straticharge System

The Heintz ram-straticharge combustion system utilizes a divided combustion chamber for charge stratification. Figure I-54 shows the precombustion chamber attached to the main combustion chamber. The prechamber contains a small air valve, a fuel-metering tube, and a spark plug. During idling and low-load operation, fuel is fed into the vertical air intake. During the suction stroke the small valve opens and admits air in a swirling motion to mix with the fuel. The mixture rotates adjacent to the spark plug and on ignition a turbulent flame is discharged into the main combustion chamber. At higher power demands, fuel is added to the unthrottled air entering through the normal air-intake valve. The Heintz system has been applied to a 4-stroke-cycle engine. The developers contend that this system will (a) reduce exhaust hydrocarbons, (b) achieve fuel economy 10 to 15 percent greater than the conventional spark-ignition engine, and (c) provide multifuel capability.

Schlamann Stratified-Charge System

The Schlamann stratified-charge system, illustrated in Fig. I-55, utilizes an auxiliary chamber for charge stratification. The auxiliary chamber contains a fuel injector and a spark plug. The main combustion chamber is contained within the piston crown, where fuel is injected by the main fuel nozzle. The charge, ignited in the auxiliary chamber, rushes through a small orifice to ignite the fuel in the main combustion chamber. The Schlamann system has demonstrated operation that was insensitive to fuel octane number, but part-load thermal efficiency was only slightly better than with the conventional combustion system, and full-load efficiency was less than that of a conventional engine. The engine operating on this system experienced severe combustion roughness.

IFP-Renault System

The IFP-Renault system, shown in Fig. I-56, employs heterogeneous carburetion. A lean mixture is fed into the combustion chamber by a carburetor. A timed rich mixture is fed through a tube into the intake port. This system is said to induce stratification to ensure ignition of the entire fuel supplied. Below 50 percent load, output is controlled by throttling as in a conventional engine. Above 50 percent load, regulation of the engine is obtained by controlling the fuel flow. Approximately 5 percent lower fuel consumption, at mid-power range, was achieved with this system. The power output was equivalent to that of a conventional engine.

Nilov Hybrid System

The Nilov hybrid engine, shown in Fig. I-57, is a Russian development that utilizes a precombustion chamber with a spark plug. A rich throttled carbureted fuel mixture is ignited in the prechamber and is directed into the main combustion chamber, which is supplied with a throttled lean mixture. The mixture in the main chamber can vary from air alone to a homogeneous air-fuel mixture. The precombustion chamber contains a stoichiometric or leaner mixture at the time of ignition.

Walker Stratofire System

The Walker Stratofire system was an attempt to develop a hybrid system that would be adapted with a minimum of modification to conventional spark-ignition gasoline engines. The system, shown in Figs. I-58 and I-59, could be inserted into the spark-plug opening in the engine. The system consists of a prechamber unit with a spark plug located in the upper end of the assembly and an automatic inlet valve located near the spark plug. The secondary carburetor is used in addition to the existing primary carburetor.

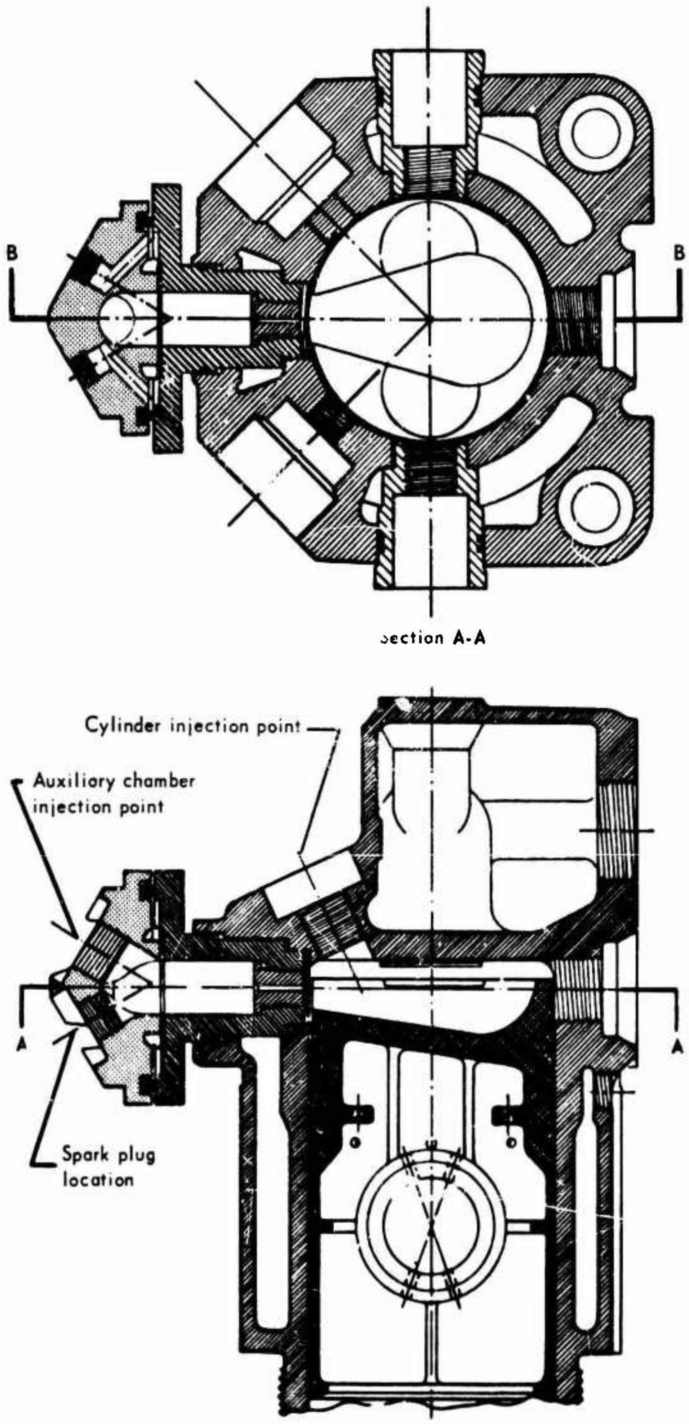


Fig. I-55—Schlamann Stratified-Charge Combustion System

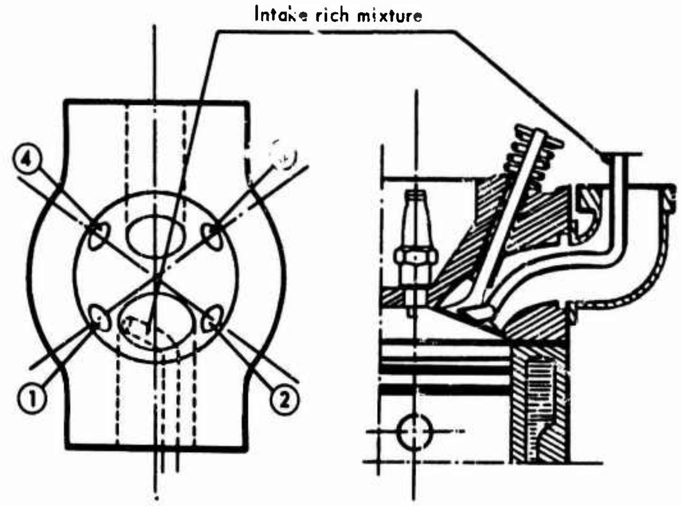


Fig. 1-56—IFP Renault Combustion System

(Spark location at Position 1 demonstrated best results.)

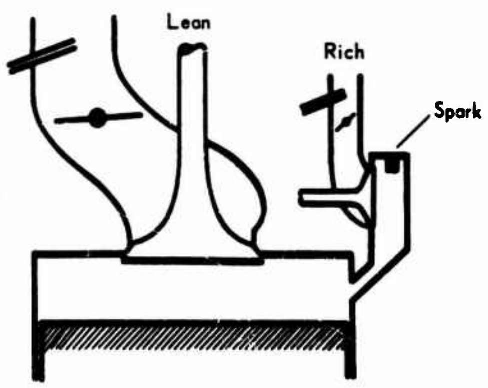


Fig. 1-57—Nilov Hybrid System

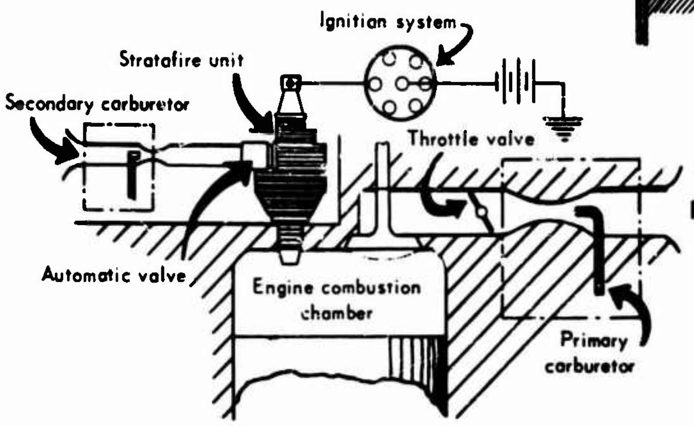


Fig. 1-58—Walker Stratofire System

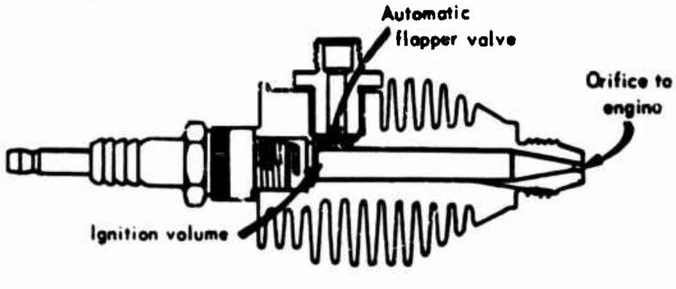


Fig. 1-59—Cross Section of Walker Stratofire Unit

A relatively rich fuel mixture is introduced from a small secondary carburetor through an automatic inlet valve into the stratofire unit and then into the engine cylinder. A lean mixture is introduced by the primary carburetor and through the engine cylinder. The stratification would then take place in the following manner: During the compression stroke the relatively rich mixture would be compressed up to and around the spark plug in the stratofire unit, so that when the spark ignited the adjacent mixture a flame front would progress through the small volume of the richer mixture, through the small antechamber, raising its pressure, and eject the burning mixture violently into the main combustion chamber of the engine, supplying a multipoint ignition system to ignite the relatively lean mixture in the cylinder. The engine would continue to be controlled by the main carburetor throttle.[8] Very little stratification was achieved by this system and it is not considered a stratified charge system.

The stratofire system did improve fuel consumption by several percent over the conventional engine. However, the primary objectives of this project were to eliminate smog-forming constituents and carbon monoxide. This project has since aroused very little interest.

Heintz 2-Cycle Stratified-Charge System

The Heintz 2-cycle stratified-charge system is similar to the Dysart-Freeman system (Fig. I-53). This system operates by controlling air and metering fuel into a prechamber. Controlled air-scavenging is also employed.

Air-Cutoff System

The air-cutoff system is an attempt to combine electric ignition in an engine system with low pumping losses. This is accomplished by using a rotary inlet valve, as shown in Fig. I-60, upstream of the conventional poppet-type

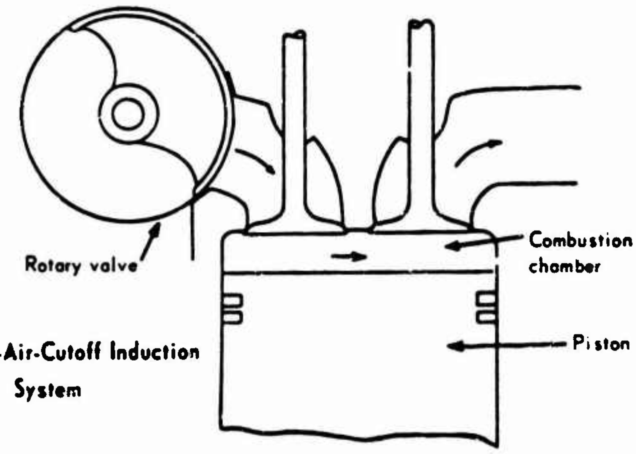


Fig. I-60—Air-Cutoff Induction System

inlet valve. A rotary valve is located at each cylinder and eliminates the conventional butterfly valve in the induction system. The rotary valve is either wide open or completely closed, and the amount of air drawn into the cylinder is controlled by the time the rotary valve is open rather than by the degree the butterfly valve is open as in a conventional system. The efficiency losses associated with pumping air through a partly closed or partly open valve are eliminated because the rotary valve is always wide open when air is being inducted into the cylinder. The air-cutoff system is expected to offer an improvement of 15 to 30 percent in overall fuel economy due to the elimination of engine pumping losses. This system is presently being investigated by ATAC.

Borg-Warner Combustion System

The Borg-Warner combustion system utilizes electric ignition, fuel-injection, and a piston with a pocket in its head that acts as a combustion chamber. This system is an attempt to achieve lean overall combustion. Fuel is injected into the combustion chamber pocket and swirl-forms a combustible mixture, irrespective of load condition. A spark plug projects into the pocket at top-dead-center igniting the mixture, which then propagates through the remainder of the lean charge causing it to burn evenly. This system is similar to the hypercycle combustion system used in the LD-465 compression-ignition military engines except that initial ignition is by electrical spark rather than by the heat of compression. This system is currently being investigated by ATAC.

Broderson Combustion System

The Broderson system utilizes a precombustion chamber with the air-intake valve and spark plug located within it. A schematic diagram of this system is shown in Fig. I-61, and a detail of the prechamber is shown in Fig. I-62. Stratification is accomplished by the change in direction of air flow as the piston passes through bottom-dead-center (bdc). The fuel charge, injected before this point, is carried into the main combustion chamber by the inducted air and is mixed with the air in the main chamber. Fuel injection continues to occur as the piston passes through bdc. The fuel injected after the piston has passed through bdc is retained in the prechamber by the air-flow reversal caused by the upward stroke of the piston, forms a combustible mixture in the region of the electric spark, and is ignited. The lean mixture in the main combustion chamber is burned by the blast of flame and hot combustion products from the prechamber.

By varying the beginning and duration of fuel injection it is possible to produce any desired mixture ratio in each chamber. This system operates at very lean mixtures of 40 to 1 (and higher) at part-load operation. Some throttling may be necessary at low power loads to avoid an excessively lean unignitable mixture in the prechamber.

Test results obtained by the University of Rochester have indicated lower fuel consumption and higher thermal efficiencies than those for a conventional spark-ignition gasoline engine. This system is currently being investigated by ATAC.

Variable-Compression-Ratio Hybrid System

This system is basically any practical hybrid combustion system incorporating the VCR piston system. An optimum compression ratio can be maintained from full load to part load with the VCR piston system, which reduces

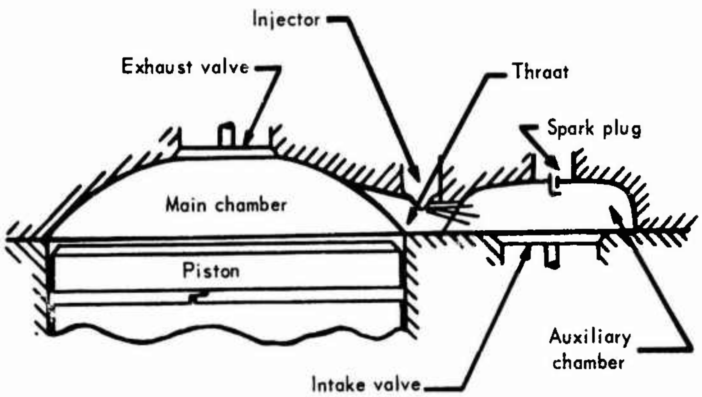


Fig. I-61—Schematic Diagram of Broderson Combustion System

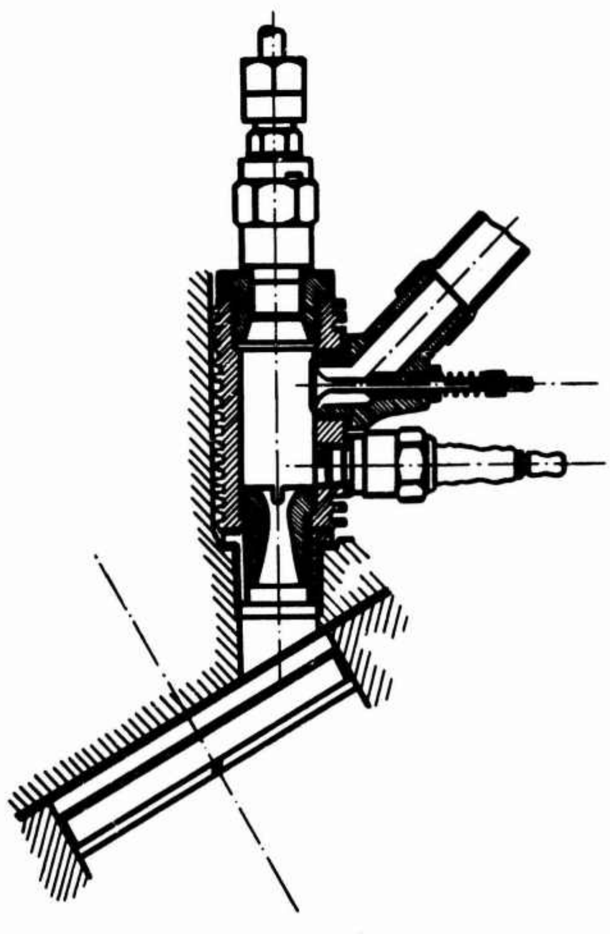


Fig. I-62—Detail of Broderson Prechamber the volume of the combustion chamber space at part-load operation. The hybrid VCR system is shown in Fig. I-63. (The VCR piston system was discussed earlier in this section.) A significant improvement (20 to 30 percent) in part-load fuel economy can be achieved since efficiency is directly proportional to the effective compression ratio in this cycle. ATAC is presently investigating this system in conjunction with a torch combustion system. The torch system utilizes a precombustion chamber to initiate combustion by electric spark. The burning gases are discharged from this chamber like a torch to ignite a lean mixture in the main combustion chamber.

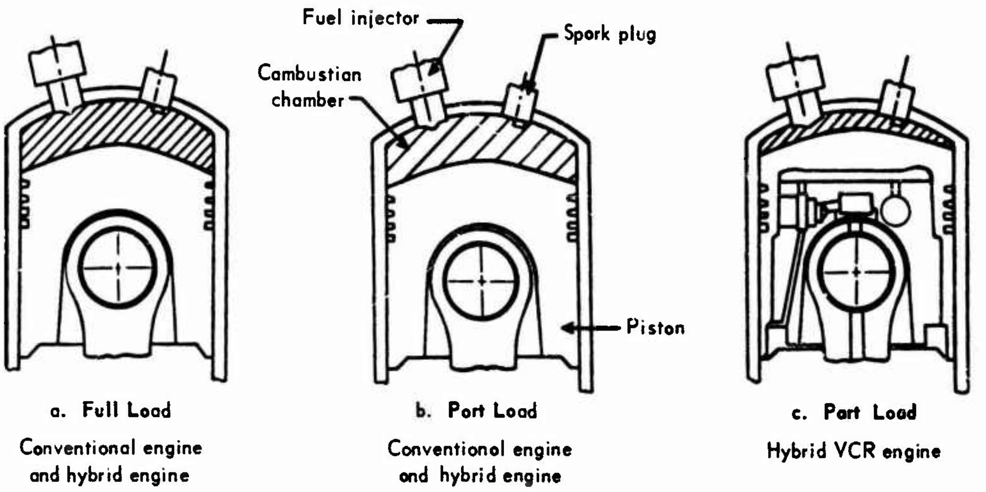
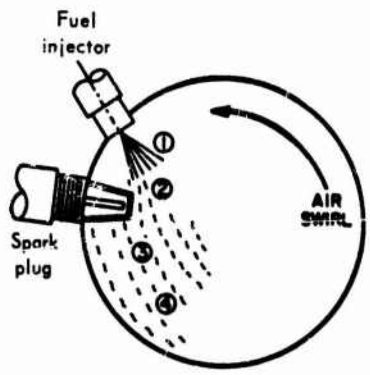


Fig. I-63—VCR Piston System Applied to Hybrid Engine

Fig. I-64—Texaco Combustion Process

① Fuel spray
② Fuel air zone
③ Flame front
④ Combustion products



Texaco Combustion Process

The Texaco combustion process (TCP) is a well-known approach to hybrid operation combining electric ignition, fuel-injection, and very lean air-fuel mixtures. The TCP, illustrated in Fig. I-64, is a stratified-charge concept where the air swirl is induced by either a shrouded intake valve or swirl port. Fuel is injected tangentially in the direction of the air swirl. Ignition occurs downstream of the injector and close to the cylinder wall. The combustion chamber is a centrally located cylindrical piston-head cavity (as in Fig. I-52). The TCP operates on a stoichiometric air-fuel mixture at full-load operation and on an excessively lean (approximately 60 to 1) air-fuel mixture at part-load operation. This system has demonstrated multifuel capabilities and better fuel economy than a conventional spark-ignition gasoline engine. Engine output is regulated by the amount of injected fuel, thereby eliminating throttling losses. The power output of engines incorporating the TCP is somewhat lower than that of a conventional spark-ignition gasoline engine. This system is currently being investigated by ATAC.

Witzky Swirl-Stratified Combustion Process

The Witzky swirl-stratified combustion process (WSP) is a well-known approach to hybrid operation. This system combines electric ignition and fuel injection and creates a violent swirl to enable operation on excessively lean air-fuel mixtures. The air swirl is created by either a shrouded intake valve, as shown in Fig. I-65, or by a swirl-deflector vane, as shown in Fig. I-66. The WSP is basically an unthrottled Otto-cycle process that utilizes direct fuel injection into the cylinder during the compression stroke and/or the final portion of the intake stroke. The fuel is injected against the air swirl within the cylinder. A small rich fuel-air cloud is produced near the spark plug by the interaction of the swirling air charge and the injected fuel droplets. This interaction causes the fuel droplets to spiral into the center of the combustion chamber to form the cloud, as shown in Fig. I-67. Figure I-68 illustrates the vector forces on the individual fuel droplet injected against the swirl air-stream where[9]

d = individual droplet at time t_1
α = the angle between the axis of the injection spray and a radial line drawn to the point of injection
V_i = instantaneous velocity of injected droplet at time t_1
V_s = air-swirl velocity at location of droplet d at time t_1
V_a = vector sum of V_i and V_s, equal to the instantaneous velocity of droplet d with respect to the air swirl
D_a = drag force on droplet as a result of air-velocity vector sum V_a
D_i = drag force opposing instantaneous velocity path of droplet
D_r = acceleration force normal to instantaneous velocity path of droplet

Engine power is controlled by varying the amount of fuel injected into the cylinder. There is no attempt to control the overall air-fuel ratio within the cylinder. The only adjustment required to change power output is the rack setting of the injection pump.

The small rich cloud is electrically ignited and propagates a flame front to ignite the lean mixture in the combustion chamber. Operation is possible on extremely lean stoichiometric air-fuel mixtures of 60:1 to 100:1 with a fuel economy approaching that of the diesel engine. Figure I-69 illustrates a comparison of a homogeneous and a stratified-charge system. The stratified-charge engine operates at a higher volumetric efficiency than a conventional spark-ignition engine. Figure I-70 illustrates a volumetric efficiency increase of 13 to 14 percent over that of the conventional engine. A performance comparison of a stratified-charge, a carbureted, and a diesel engine is shown in

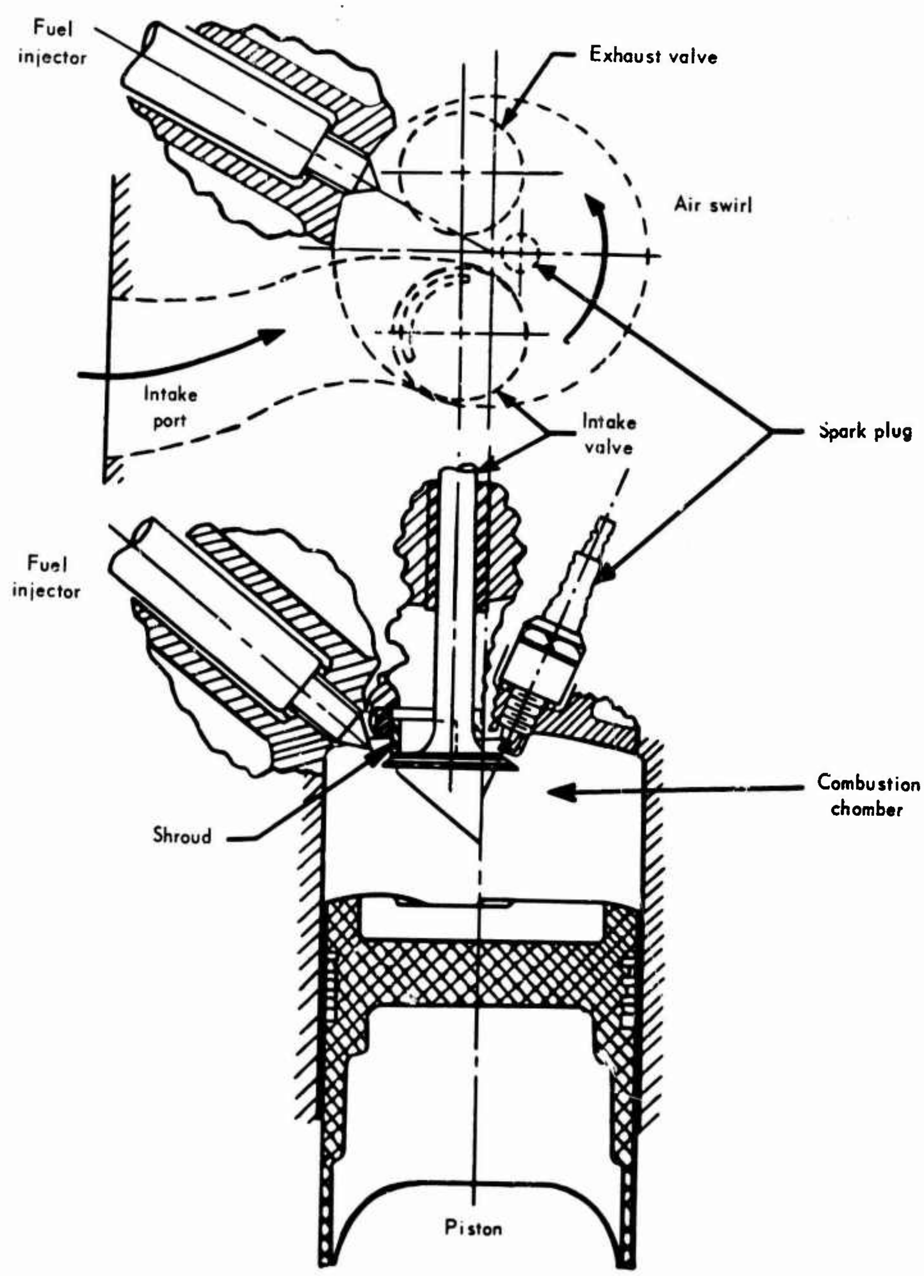
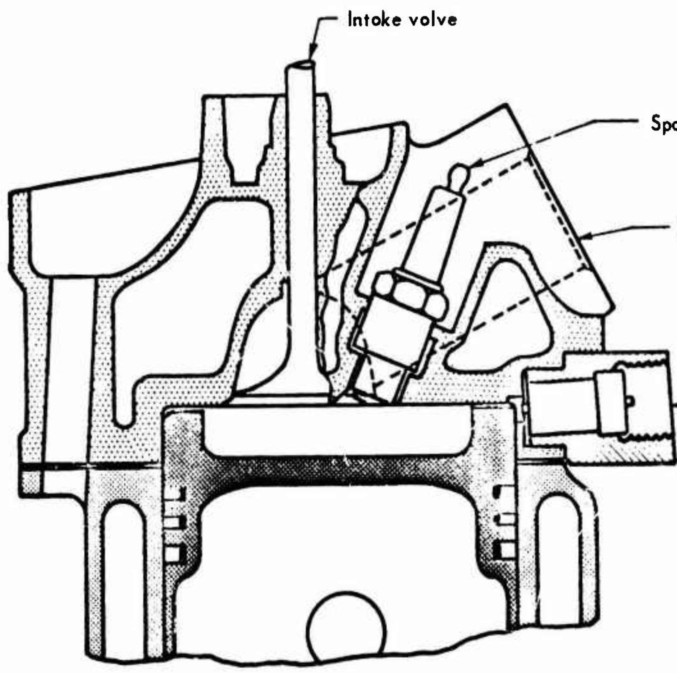


Fig. 1-65—Swirl-Stratification System Using Shrouded Intake Valve



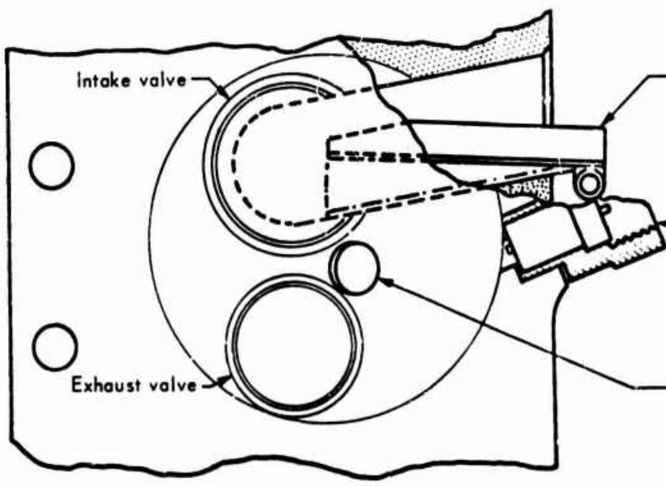


Fig. I-66—Swirl-Stratification System Using Swirl-Deflector Vane

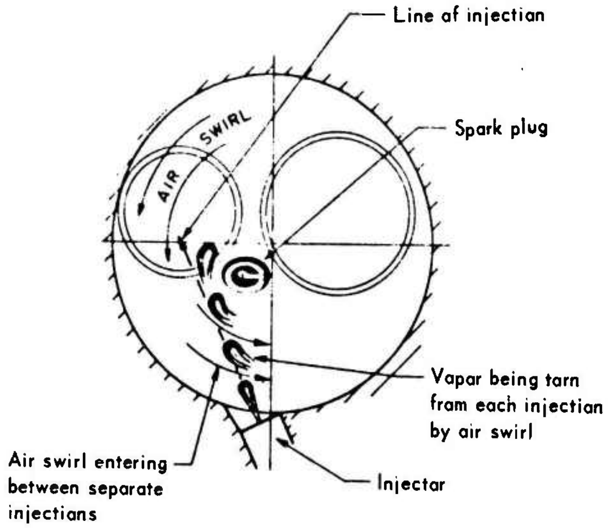


Fig. I-67—Pulsed Fuel Injection at High Power Output

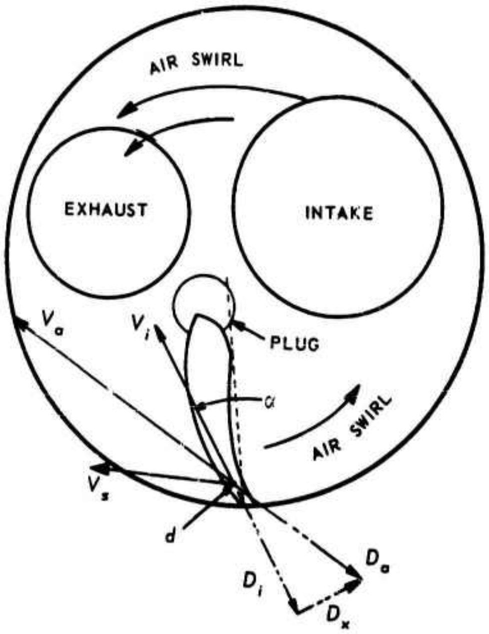


Fig. I-68—Vector Forces on Injected Fuel Droplet in Swirl-Stratified System

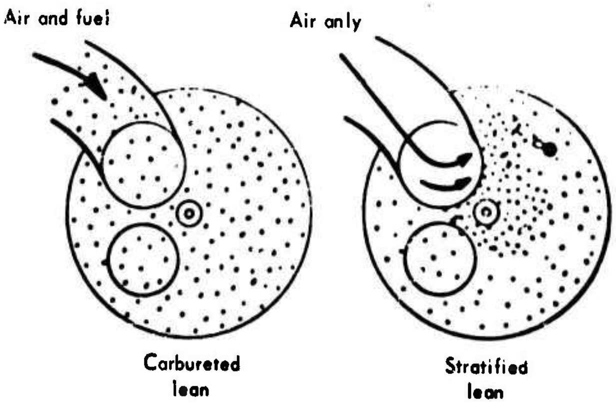


Fig. I-69—Comparison of Homogeneous and Stratified-Charge Systems

Fig. I-71. The stratified-charge engine has a peak power increase of approximately 30 percent above that of the same engine carbureted. The fuel consumption approaches that of the diesel engine. The swirl-stratified engine operates equally well on gasoline, CITE, JP-4, and diesel fuel, and engine response is quicker than that of conventional spark-ignition and diesel engines. The swirl-stratification hybrid system is presently being investigated by ATAC.

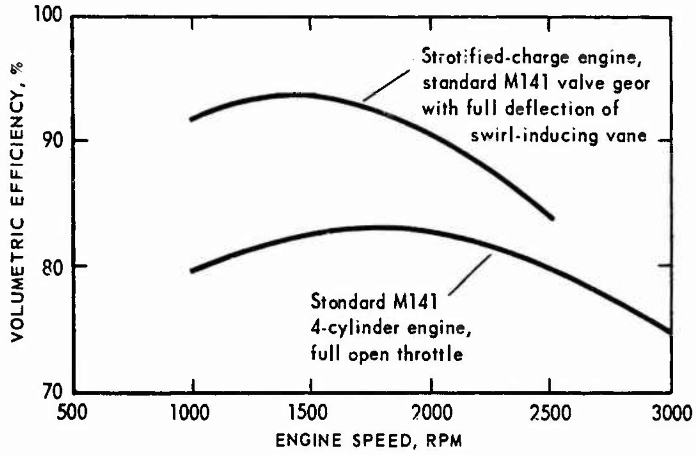


Fig. I-70—Comparison of Volumetric Efficiency of Carbureted and Stratified-Charge Engines

TECHNOLOGICAL FORECAST

The trend-forecast charts illustrate the estimated level of technological achievement (through 1980) of the spark-ignition gasoline engine (Figs. I-72 to I-74), the compression-ignition engine (Figs. I-75 to I-77), and the hybrid engine (Figs. I-78 to I-80).

CONCLUSIONS

Spark-ignition gasoline engines have been and will continue to be developed by industry for commercial applications. These engines must be modified before their acceptance for use in tactical vehicles. Past improvements in fuel economy, weight and size, and reliability have been substantial. Although the rate of future improvements will diminish in magnitude, these engines will continue to have application in both commercial and tactical vehicles for many years in the future.

Compression-ignition engines have been and will continue to be developed by industry for commercial applications in power ranges to 600 hp. These engines must be modified before their acceptance for use in tactical vehicles. Although the military has selected these engines for use in tactical vehicles, improvement of the physical and performance characteristics in power levels

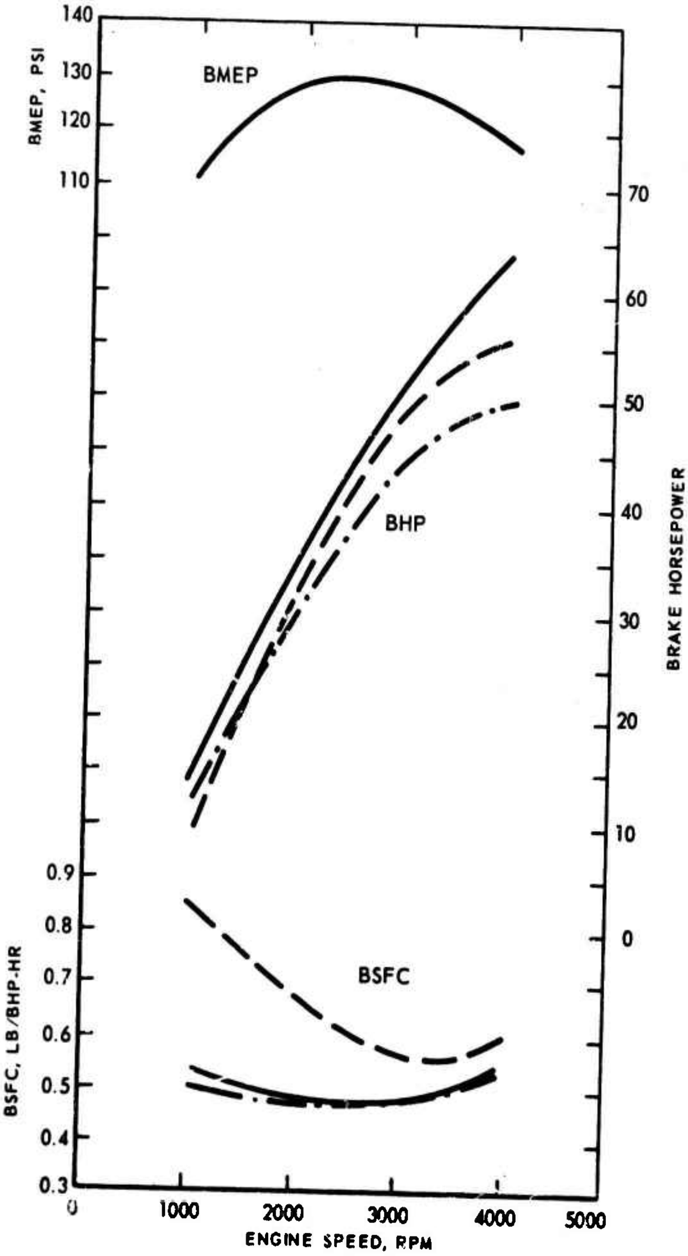


Fig. I-71—Comparison of Full-Throttle Performance of Stratified, Carbureted, and Diesel Engines

Stratified 93 octane
Carbureted 100 octane
Displacement 107.5 in.^{3}
Cylinders 4

Diesel
Displacement 115 in.^{3}
Cylinders 4

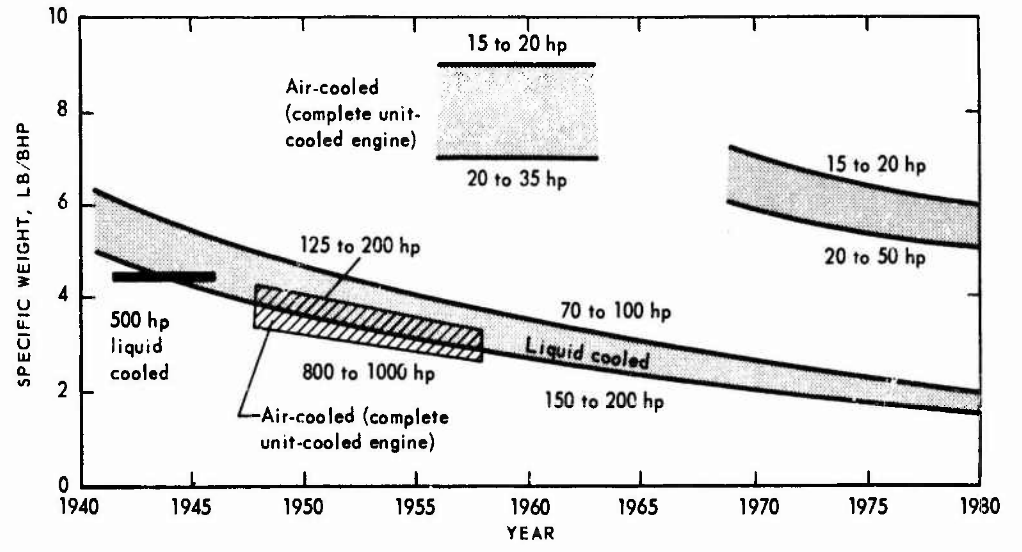


Fig. I-72—Trend Forecast of Specific Weight of Spark-Ignition Gasoline Engines

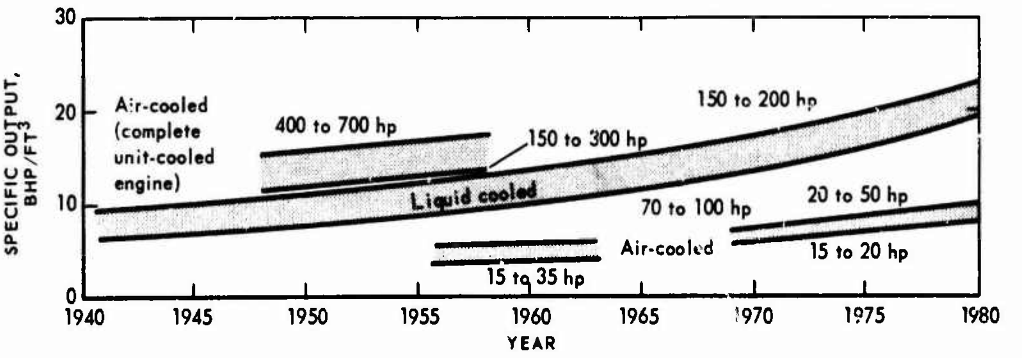


Fig. I-73—Trend Forecast of Specific Output of Spark-Ignition Gasoline Engines

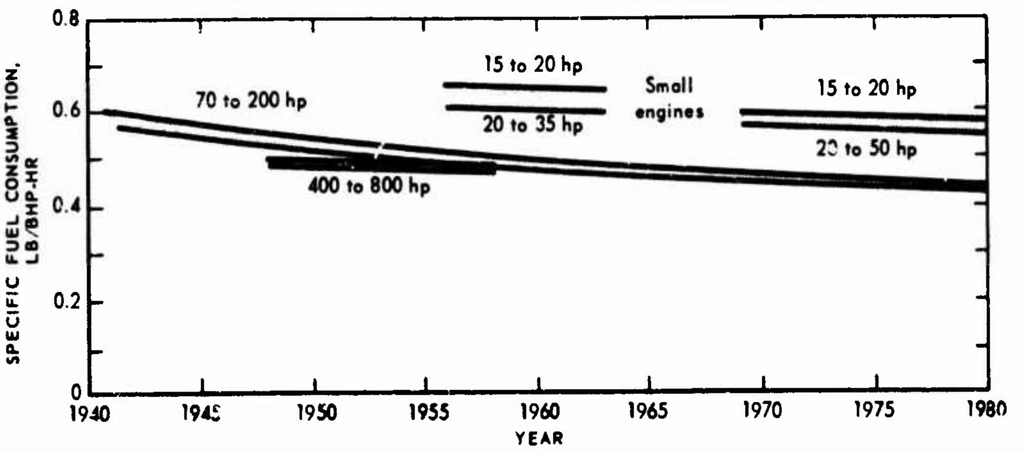


Fig. I-74—Trend Forecast of Fuel Consumption in Spark-Ignition Gasoline Engines
(Best-point data.)

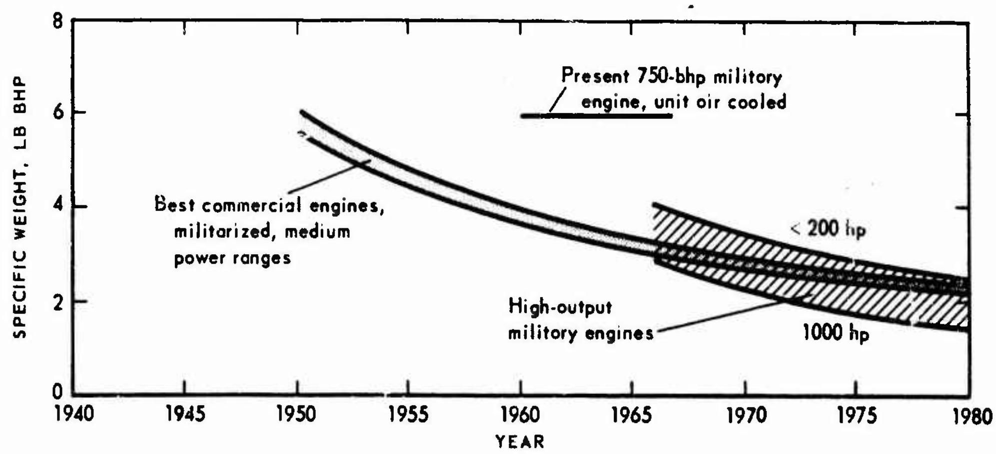


Fig. I-75—Trend Forecast of Specific Weight of Compression-Ignition Engines

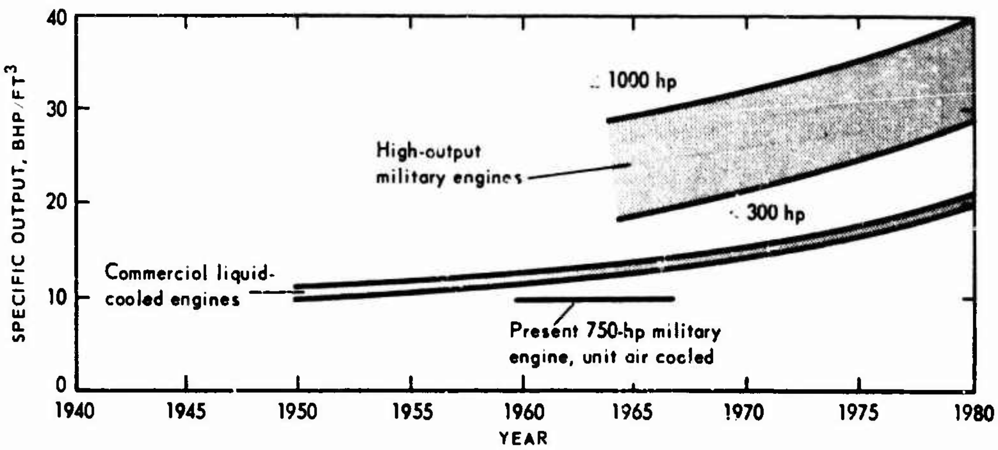


Fig. I-76—Trend Forecast of Specific Output of Compression-Ignition Engines

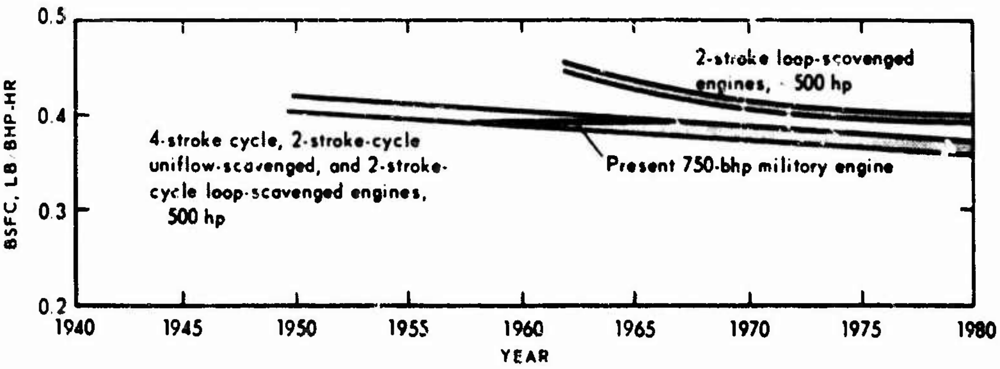


Fig. I-77—Trend Forecast of Fuel Consumption in Compression-Ignition Engines
(Best-point data.)

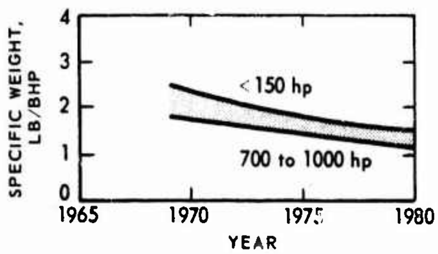


Fig. I-78—Trend Forecast of Specific Weight of Hybrid Engines

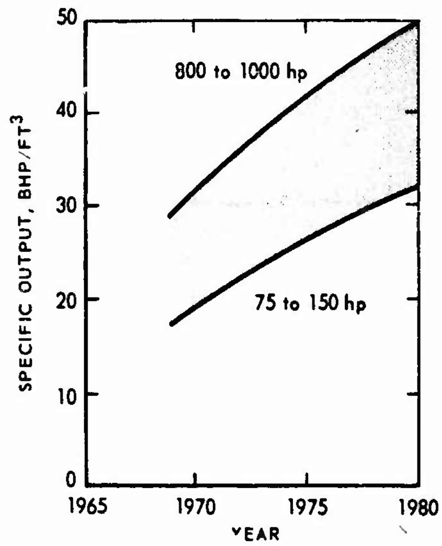


Fig. I-79—Trend Forecast of Specific Output of Hybrid Engines

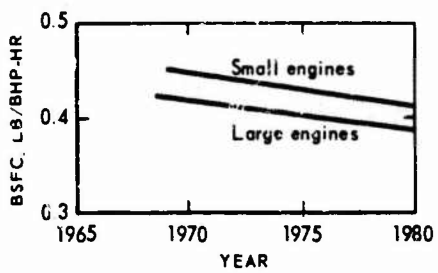


Fig. I-80—Trend Forecast of Fuel Consumption in Hybrid Engines

(Best-point data.)

currently commercially available and in the higher power levels not available is needed. To meet this need the military has sponsored the development of such engines as the hypercycle, AVM, VCR, VHO, and EHO.

The hybrid engine attempts to incorporate the best features of the spark-ignition and compression-ignition engines. Although prototype-engine test results have been encouraging, no successful hybrid engine has been developed to date. With further development, the hybrid engine promises to evolve into a multifuel power source that is highly efficient, lightweight, and compact. Although industry has incentive to develop a hybrid engine, their development efforts are limited and insufficient for successful development within the 1980 time frame. Therefore, it is concluded that Government support of an R&D program could produce a hybrid engine that would improve the physical and performance characteristics of many future tactical vehicles.

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Chapter 4

AMMONIA ENGINES

The ammonia engine, an energy-conversion device, can furnish a means for the production of mechanical work by converting potential energy from anhydrous ammonia fuel. Such an engine can operate on the principle of a turbine or a reciprocating engine and in either case can be spark- or compression-ignited. A new engine may be designed for the specific purpose of using an anhydrous ammonia fuel or available gasoline and diesel engines can be modified for the purpose.

Increased logistics problems attending the supply of petroleum products for military use during WWII created an interest in ammonia engines. Petroleum products comprised approximately 50 percent of US Army overseas shipping in WWII. The proportion of such shipping increased to about 70 percent during the Korean action.[1] Consideration of these factors led to military-initiated R&D programs for engines that could use a fuel manufactured in the field. Numerous concepts involving the use of chemicals now available were evaluated. The evaluation revealed that the best approach to attaining the goal would be the manufacture of ammonia fuel through use of a nuclear-powered energy-depot system.[1]

EXPERIMENTAL DEVELOPMENTS

From 1964 to 1966 the US Army Engineering Research and Development Laboratories (USAERDL), Ft Belvoir, Va., had an ammonia-fueled engine under test. The engine, a Chevrolet gasoline engine modified to use ammonia fuel of commercial manufacture, is installed in a pickup truck in place of the standard engine. This pickup truck is the only known vehicle in operation at present having an ammonia-fueled engine. All other engines, whether modified or designed to use ammonia fuel, are laboratory mounted and undergoing dynamometer tests.

Studies of ammonia as an engine fuel have been, and are being, conducted by the US Army and US Navy as well as by various industrial and academic research organizations. In these studies emphasis has been placed on the conversion of existing compression-ignition and spark-ignition reciprocating engines to ammonia-burning engines through use of an adaption kit. Success in engine conversion has been achieved by USAERDL, GMC, and the Continental Motors Corporation. The studies have centered about logistic vehicles.

The adaption kit, produced by the Continental Motors Corporation for the purpose of converting a standard engine to an ammonia engine, contains the following items:

(a) A new ignition system to increase spark temperature.

(b) A dissociator to isolate a small amount of hydrogen from the intake manifold. This hydrogen then serves as a combustion promoter accelerating the burning of ammonia.

(c) An ammonia pump to replace the carburetor.

(d) New heads to increase the engine compression ratio.

The incorporation of the kit items in a standard engine does not appreciably increase the size and weight of the engine.

In addition to installation of the adaption-kit items, several other modifications are required to convert an existing hydrocarbon-fueled engine to an ammonia-fueled device. Parts made of copper, brass, bronze, zinc, and galvanized metal corrode on contact with ammonia. These parts, i.e., fuel lines, fittings, and related parts, must be replaced with homogeneous iron or steel components on which the corrosive effects of ammonia are negligible. Also engine timing must be changed and the fuel tank must be modified. The modifications cited are not too difficult to perform nor are the mandatory fuel-tank changes difficult.

It is to be noted that converted engines, or engines designed to use ammonia fuel, do not have a multifuel capability but rather are restricted to the use of ammonia fuel.

DISCUSSION

At present the installation of an ammonia engine, converted from a gasoline or diesel engine, reduces the operating capability of a vehicle and limits the use of such a vehicle to areas where full power is not required. Figures submitted by USAERDL indicate that gasoline engines, when converted to use ammonia fuel, suffer a horsepower loss of approximately 20 percent. A power loss of this magnitude degrades the operational capability of an engine when full power is required. The Continental Motors Corporation is attempting to reduce this loss without supercharging the converted engines.

To achieve equal driving distance in a vehicle having a converted gasoline engine the amount of ammonia fuel required will be at least 2.8 times by volume and 2.35 times by weight greater than the amount of gasoline required. Thus it is necessary to increase the size of, and pressurize, the vehicle's fuel tanks to achieve the mileage range obtainable with a gasoline-powered engine.[2]

In theory the startup time for an ammonia-fueled engine is identical with that of a hydrocarbon-fueled engine, except in Arctic conditions where an electric heater is needed to ensure responsive starting by heating the dissociator.

A comparison of the potential energy available from ammonia fuel with that available from certain other fuels is shown in Table I-9.[1]

Ammonia Leaks

Ammonia gas emanating from an ammonia fuel tank or engine in small concentrations is not expected to result in extensive physical damage to personnel. The effects of ammonia gas in large concentrations are not unlike those produced by tear gas. The pungent odor of ammonia gas gives adequate warning of its presence, and it is unlikely that anyone would remain in the vicinity of an ammonia-gas leak unmasked unless he were trapped.

TABLE I-9

Comparison of Ammonia Fuel with Certain Other Fuels

Fuel	Heating value, Btu/lb	Specific density, lb/ft^3
Ammonia liquid[a] (70°F, 125 psia)	7,492	42.6
Diesel fuel	18,558	52.01
Gasoline	18,700	48.6
Hydrogen gas (70°F, 2000 psia)	51,593	0.0065
Hydrogen liquid (−423°F, 15 psia)	49,150	4.44
Hydrazine liquid (70°F, 15 psia)	6,723	62.8

[a]Liquid ammonia (fuel) must be stored in pressurized, cylindrical fuel tanks that cannot be filled to capacity because the expansion of ammonia produces hydrostatic pressures that may burst the tank.[3]

TABLE I-10

Gaseous Ammonia Safety Levels[3]

Effect	Ammonia concentration, ppm of air, (by volume)
Minimum detectable odor	53
Maximum concentration tolerable for prolonged exposure (8 hr)	100
Maximum concentration tolerable for short-time exposure (½ to 1 hr)	300 to 500
Minimum amount to cause immediate eye irritation	698
Minimum amount to cause coughing	1720
Dangerous for short-time exposure (½ hr)	2500 to 4500
Fatal for short-time exposure (rapid action)	5000 to 10,000

The extent of damage to living subjects that can be expected from the concentration of ammonia gas in air is a function of the density of the concentration and the duration of the subject's exposure. The maximum tolerable concentration of gaseous ammonia in air to which living persons or animals may be exposed for an 8-hr period is 100 ppm (see Table I-10[3]). Under identical conditions the maximum permissible concentration of gasoline is from 500 to 1000 ppm.[4] The danger of an explosion from ammonia in the air is not as great as the explosive potential presented by gasoline. The explosive limits for ammonia are from 16 to 25 percent ammonia per 84 to 75 percent (by volume) of air, and the explosive limits for gasoline are from 1.3 percent to 6 percent gasoline (by volume) per 98.7 to 94 percent of air.[4]

The maintenance of an engine converted to use ammonia fuel should not differ from that of a standard internal-combustion engine, but regular overall maintenance is essential owing to the potential hazards to personnel inherent in ammonia-fuel leaks. Strict adherence to safety precautions is required of persons operating or servicing a vehicle having an ammonia-fueled engine. However, total compliance with safety requirements cannot compensate for leaks resulting from system exposure and vulnerability during combat conditions.

Ammonia Exhaust

The hazardous content of ammonia-engine exhaust gases is nitric oxide. The concentration of nitric oxide accumulating during ammonia-engine operation is substantially less than that of the detrimental exhaust gases produced by hydrocarbon-fueled engines under similar conditions and has a lesser adverse effect on personnel.

Nuclear-Powered Energy Depot

A study of the nuclear-powered energy depot concept is in progress at the Allison Division of GMC and at the Allis-Chalmers Manufacturing Company. These studies are being conducted under the sponsorship of the US Army.

A nuclear-powered energy depot, situated in a strategic location, could be sustained by a single energy unit and produce approximately 120 gal of ammonia fuel each hour. This rate of productivity at a field depot could relieve logistic supply lines of vast petroleum shipments.

The location and operation of a nuclear-powered energy depot for ammonia production depends on an abundant supply of fresh or salt water. After designation of an appropriate location, a single energy unit (a nuclear reactor) could be transported by air to the site and an energy depot established close to water. At this depot the nuclear reactor would dissociate hydrogen from water and fractionate nitrogen from air. The two elements then would be combined through a chemical process to make ammonia fuel.

Use of ammonia engines presupposes the adoption of the nuclear-powered depot concept. To relieve logistic supply lines of the shipment of hydrocarbon fuels the installation of an energy depot might well be warranted. A specific example of a place where an application of the nuclear-powered energy depot concept could prove most advantageous, not in the province of US military activity, is Rhodesia. If the Rhodesians, deprived of hydrocarbon fuels during the 1966 boycott, had had an energy depot for ammonia-fuel production situated adjacent to the required water supply, existing engines could have been modified to permit utilization of the fuel produced. This action then would have precluded the possibility of Rhodesian vehicle immobilization. The example grants the Rhodesians the technological capability and the components necessary for the modification of available engines.

To give an example in the military sphere, application of the energy-depot concept could offer definite advantages in Arctic operations where supply-line transportation is all but impossible for most of the year.

EVALUATION

Limitations and Advantages

Ammonia engines possess inherent drawbacks that at this time prevent full utilization of this vehicular power source and the development of its apparent potentials. The most significant drawbacks deterring immediate attention to this engine include hazards to personnel, engine power loss, and the presentation of logistics problems now existent. A major advantage is that any standard engine in use today in wheeled, tracked, or special-purpose vehicles can be successfully modified to use ammonia fuel.

The ammonia engine's lack of a multifuel capability is offset inasmuch as use of this engine (in conjunction with a nuclear-powered energy depot) could effect a great reduction in the transportation requirements for hydrocarbon fuels. However, hydrocarbon fuels would be available for issue to those areas where use of an energy depot would not be practical.

Ammonia-Engine Costs

Ammonia-engine costs have not as yet been ascertained pending a firm engine design. However, it is anticipated that the costs of an ammonia-fueled engine, once designed and in production, will vary little from known, standard engine costs.

Energy-Depot and Fuel Costs

USAERDL is compiling cost figures on an ammonia-producing nuclear-powered energy depot. These cost figures will include the initial cost of the depot facility and the production costs of ammonia. A comparison then can be made of the ammonia-fuel production costs with hydrocarbon-fuel costs.

Manufacturers producing ammonia for commercial use have stated that anhydrous ammonia, at present, is produced in limited quantities. The belief was expressed that if the demanded quantities of this item were large enough to warrant gearing the processing effort to mass production, conceivably the price of this fuel could be brought down to a figure as low as $15 per ton. Ammonia fuel costs then would be approximately 4 cents per gallon or 12 cents for an energy source equivalent to a gallon of gasoline.

Hydrocarbon fuels can be purchased by the military at most depots throughout the world for about 13 cents per gallon. However, in areas where logistics problems are extreme the cost of hydrocarbon fuels to the user may soar to a figure as great as $5 per gallon owing to transportation and handling costs incurred before receipt of the fuel by the user.

CONCLUSIONS

Situations conducive to the installation of a nuclear-powered energy depot are those where:

(a) Sustaining a scheduled supply line of all fuels is practically impossible. (In certain areas, climatic conditions are such that fuel can be delivered only during specific seasons or during limited periods of each year.)

(b) A very high price is paid for fuel.

Two new logistics problems become apparent with the introduction of ammonia engines for tactical vehicles.

(a) Despite the fact that the logistics burden of hydrocarbon fuel transportation would be decreased, the number of vehicles needed would increase because ammonia-fueled vehicles cannot be used under combat conditions. Therefore two types of vehicle for the same horsepower class would be required, one for combat use and one for logistics purposes.

(b) An entire new line of some spare parts would be required. The military would have to incorporate many new parts in the logistics system, supply depots would have to accommodate and implement issue of the new ammonia-fueled engines, and maintenance depots would have to gear for their repair.

At the selected installation sites ammonia-producing energy depots could provide heat and electrical power for the facility. Ammonia produced could also be used as fuel for special-purpose vehicles including fork-lift trucks, cranes, and other depot and engineering-type vehicles.

It is predicted that after a period of sustained functioning, the operation of several ammonia-producing energy depots will aid in furthering technological advances. These advances will include:

(a) A great reduction in ammonia-engine vehicle hazards.

(b) Increased ammonia-engine efficiency.

(c) The possible development of a successful variable-compression-ratio engine that will burn various hydrocarbon fuels as well as ammonia.

Based on the foregoing analysis it is apparent that R&D on the nuclear-powered energy depot concept for ammonia-fuel production should be continued. At the onset this effort should be directed toward establishing energy depots at large military installations that have a relative security from attack by hostile forces, have an available water supply, and are located, in general, at what might be termed "strategic sites." The successful use of the initial ammonia-producing energy depots and the progressive technological advancements may result in the use of ammonia fuel for many types of military vehicles. However, it is concluded that R&D programs for ammonia-fueled engines for use in tactical vehicles are not warranted at this time.

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Chapter 5

ROTARY-PISTON ENGINES

INTRODUCTION

The rotary-piston engine, or rotating combustion engine, is a recent development of an old unconventional engine-design concept. In past years many internal-combustion rotary-engine designs were conceived and patented. The objectives of the early inventors were to develop an engine with power-producing components rotating about a central axis, thereby eliminating the reciprocating elements and the undesirable reciprocating forces inherent in conventional reciprocating engines. However, none of the rotary engines displayed any lasting prominence owing to their failure to achieve performance characteristics equal to those of the conventional reciprocating-piston engine. In 1951 a German engineer, Felix Wankel, conducted an investigation of all known rotary engines. His study concluded[1] that the major problem areas that prevented the rotary engine from achieving success were:

(a) Multiplicity of cycles and arrangements.

(b) Sealing of the combustion-chamber walls and the power-producing rotating piston.

(c) Development of a proper thermodynamic and gas cycle with adequate port areas and timing of cycle events.

WANKEL-TYPE ROTARY-PISTON ENGINE

Through experimentation on test engines, Wankel was able to reduce most of the problems to what he considered an acceptable level. At this time the rotary-piston engine showed enough promise to be worthy of further development.

World patent rights to the Wankel-type rotary-piston engine are held by two West German concerns, NSU Motorenwerke, A. G., and Wankel, G.M.B.H. Intensive R&D work has centered about this engine concept because these two concerns have negotiated license agreements with some 15 well-known companies throughout the world. The Curtiss-Wright Corporation holds exclusive rights for this power source on the North American continent.

The rotary-piston engine concept is simple, lightweight, and compact and has the possibility of being mass-produced at a lower cost than conventional reciprocating gasoline engines. The engine concept can be adapted to spark-ignition, compression-ignition, and hybrid combustion principles. It can be designed and built to be either air- or liquid-cooled.

Basic Design

The basic design concept consists of two main components, as shown in Fig. I-81. One member (3) is an outer, stationary center housing with two attached parallel side housings. The inner surface of the center housing has the configuration of a two-lobe epitrochoid. The other member (1) is a trochoidal inner rotor that rotates relative to the outer member. The inner rotor is placed eccentrically on a shaft centered within the outer housing. During rotation the inner rotor corners approximate the epitrochoidal shape of the outer housing, as shown in Fig. I-82. The rotor is inner-planetary-geared to a stationary gear affixed to the end housing and is contoured so that all three apex points are in contact with the outer housing at all times. The output shaft, centrally located in the epitrochoid, is driven by the inner rotor through the eccentric lobe of the output shaft. The contained volume between each rotor side and the rotor housing varies sinusoidally, giving a difference between minimum and maximum chamber volume according to the position of the rotor (Fig. I-82). The minimum and maximum positions are equivalent to top-dead-center and bdc and the volume swept out by the rotor in going from one to the other is defined as the "sweep volume" or displacement. The inner rotor provides three power strokes per revolution, and the engine output shaft rotates three times for each rotor revolution. Figure I-83 illustrates the combustion cycle of a gasoline rotary-piston engine that operates on the standard 4-stroke Otto-cycle principle using spark ignition and carbureted fuel induction.

The Curtiss-Wright rotary-piston engine contains no conventional pistons, connecting rods, valves, or valve mechanisms. The engine aspirates and exhausts through porting located in either the outer periphery of the rotor housing or in the side plates. Present engines have side intake porting and peripheral exhaust porting. The rotary-piston engine, unhampered by piston or valve-gear inertia, dynamic forces, and couples due to unbalance or breathing restrictions, has an inherently high speed capability.[2] Because of the absence of

TABLE I-11

Comparison of Number of Parts in Rotary-Piston Engine with Those of a Conventional Automotive V-8 Engine

Category	RC2-60-U5 engine	Conventional automotive V-8 engine
Horsepower	185	195
Total number of parts	633	1029
Number of moving parts in power section and drive line	154	388

these parts (see Table I-11), a typical rotary-piston engine contains approximately 38 percent fewer total parts and 60 percent fewer moving parts than a comparable reciprocating automotive V-8 engine. Approximately 90 percent of the total parts would be common to a family of multirotor engines.[3]

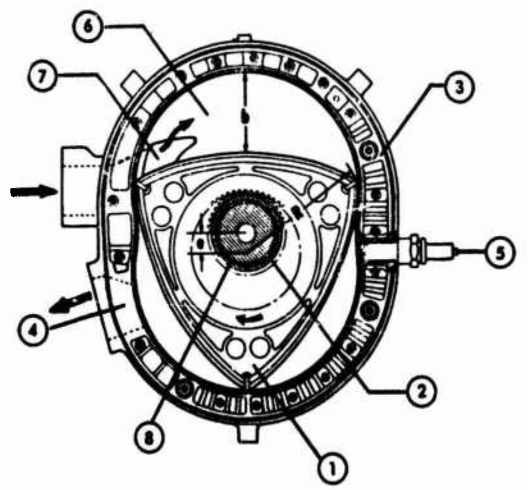


Fig. I-81—Cross Section of Rotory-Piston Engine

1. Rotor with internal rotor gear
2. Stationary gear
3. Rotor housing
4. Exhaust port
5. Spark plug
6. Side housing
7. Intake port
8. Main bearing

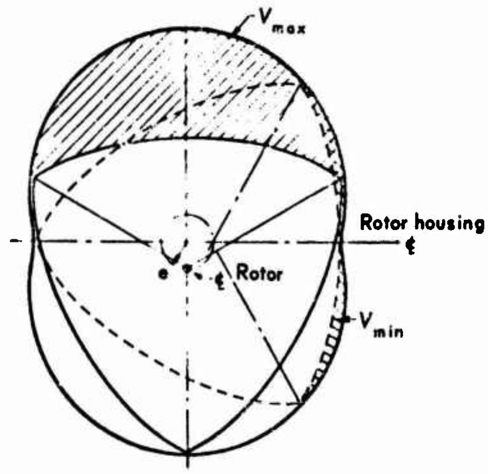


Fig. I-82—Rotary-Piston Engine Geometry

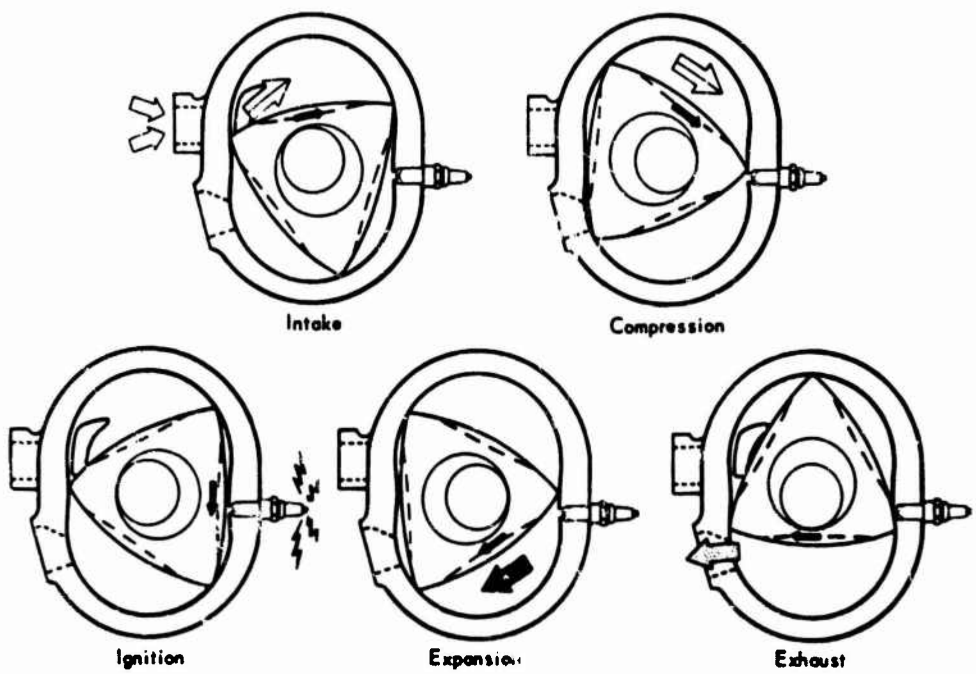


Fig. I-83—Combustion Cycle of Rotary-Piston Engine

Discussion

The Curtiss-Wright Corporation has developed and built a number of engines in 1-, 2-, and 4-rotor configurations (see Figs. I-84 and I-85). These engines incorporate a rotor of 60-in.^{3} displacement and burn gasoline, utilizing spark ignition and carbureted fuel induction. Engines can be constructed with 1, 2, 3, 4, or 6 rotors. However, it appears that stacking of more than 6 rotors would be impractical owing to the long crankshaft required, and that a 5-rotor engine would suffer from serious imbalance conditions.

Rotors may also be constructed to meet any displacement or power requirement, such as 150- or 308-in.^{3} displacement, but the advantage of family capabilities would be lost. Table I-12 illustrates the family capabilities of

TABLE I-12

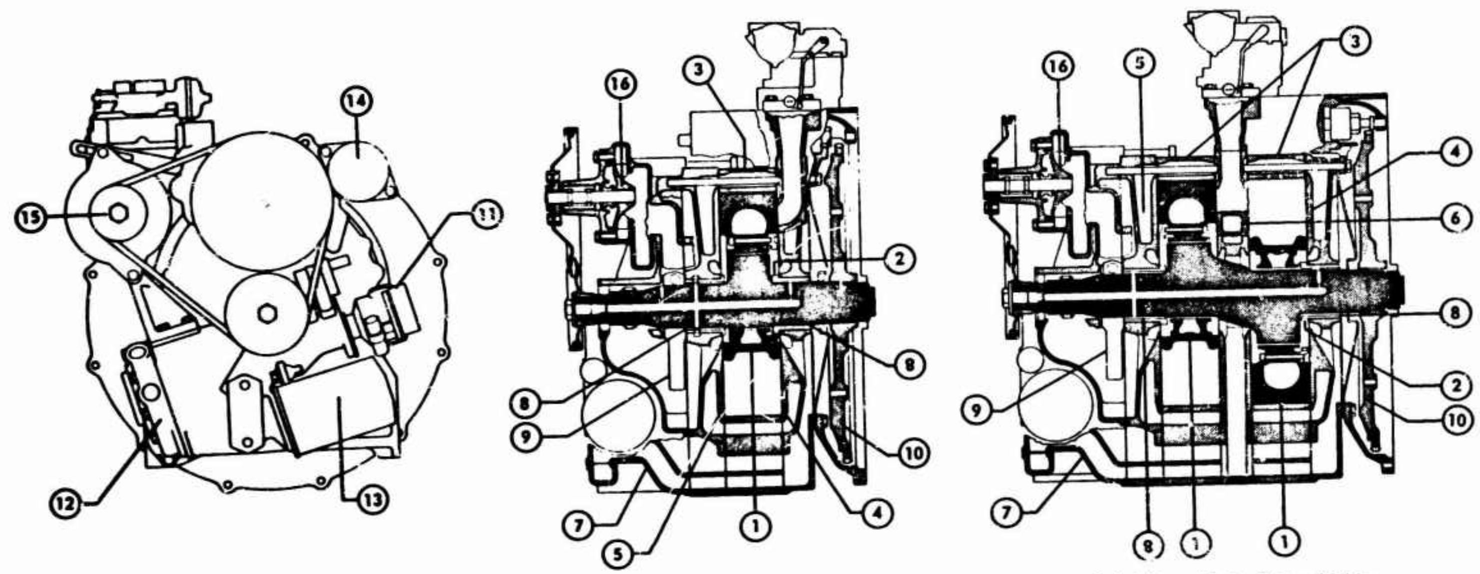
Family Capabilities of Rotary-Piston Engines, Spark-Ignition, Gasoline Fuel

(Based on present automotive rating of two-rotor engine RC2-60-U5)

No. of rotors (60-in.^{3} displacement rotor)	Rating, hp	
	Automotive[a]	Military[b]
One rotor	92	80
One rotor, turbocharged[c]	115	100
Two rotors	135	160
Two rotors, turbocharged[c]	230	200
Three rotors	277	240
Three rotors, turbocharged[c]	345	300
Four rotors	370	320
Four rotors, turbocharged[c]	460	400
Six rotors	555	480
Six rotors, turbocharged[c]	690	600

[a]Curtiss-Wright automotive-industry rating.
[b]Estimated military durability rating.
[c]Estimated; (horsepower based on assumption of 25 percent increase from turbosupercharging).

rotary-piston engines with rotor displacements of 60 and 90 in.^{3}. The table lists both automotive-industry ratings and estimated military durability ratings for comparison purposes and is based on present-day technology. Although the rotary-piston engine is an inherently higher-rpm engine than the conventional reciprocating engine, it is believed that it too must be derated in horsepower output for military usage, much the same as a conventional piston engine is derated from automotive-industry ratings. The severity of military vehicle operations dictates that the engine must operate at high-power loadings (as compared to automobiles, whose engines operate at low to moderate loadings) for sustained periods of time. Therefore rotary-piston engines must be more conservatively rated for military use than for automotive use to assure their durability. Figure I-86 illustrates the family capabilities of the rotary-piston



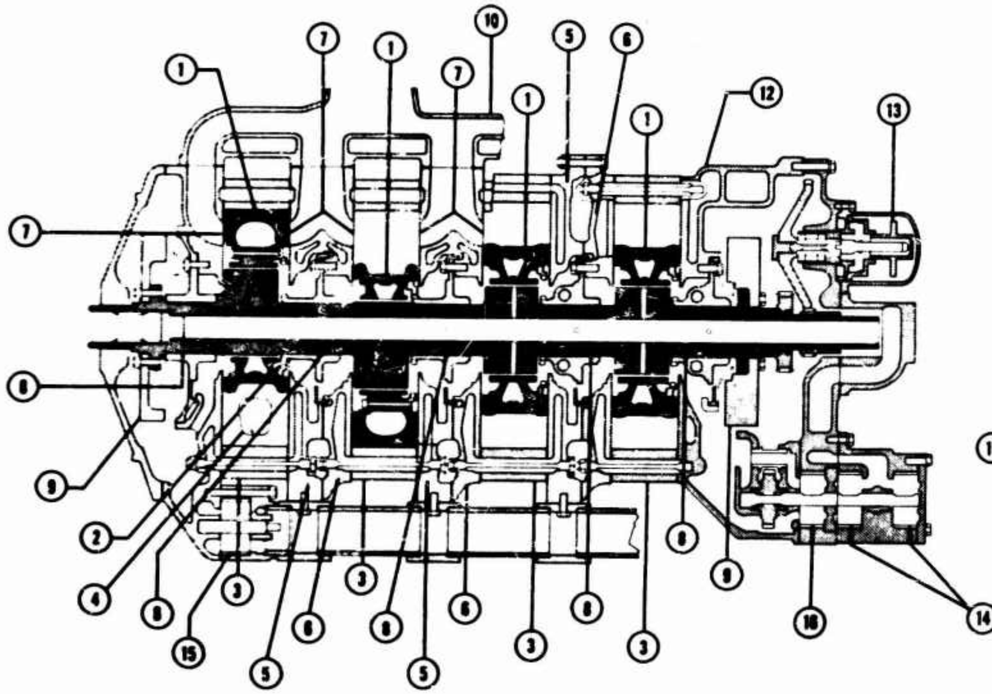
a. Front View

b. Side View, Single Rotor 92 hp

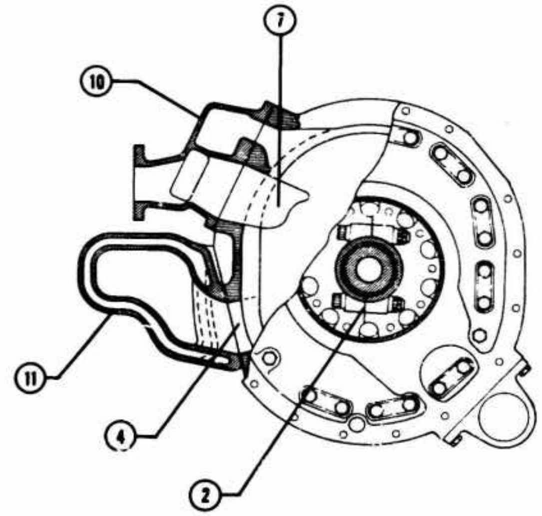
c. Side View, Twin Rotor 185 hp

Fig. I-84—Rotary-Piston Engine Configurations, 1- and 2-Rotor Engines

1. Rotor
2. Stationary gear
3. Rotor housing
4. Side housing—drive side
5. Side housing—antidrive side
6. Intermediate housing
7. Accessary housing
8. Main bearing
9. Balance weight
10. Flywheel cum balance weight
11. Ignition contact maker
12. Oil pumps
13. Oil filter
14. Starter
15. Generator
16. Water pump



a. Side View



b. End View

Fig. I-85—Rotary-Piston Engine Configuration, 4-Rotor Engine

4RC6 engine cross sections.

1. Rotor
2. Split stationary gear with main bearing
3. Rotor housing
4. Exhaust port
5. Side housing—drive side
6. Side housing—antidrive side
7. Intake port (dual intake)
8. Main bearing
9. Flywheel cum balance weight
10. Intake manifold
11. Exhaust manifold
12. Accessory gear box housing
13. Ignition contact makers
14. Oil pressure pumps
15. Oil scavenge pump—front
16. Oil scavenge pump—rear

engine with rotors of 60-, 90-, and 150-in.^{3} displacement. These capabilities are based on present-day technology. Supercharging or turbocharging has not yet been applied to the rotary-piston engine, but there appears to be no reason why it cannot be applied to these engines. Application of turbosupercharging should boost the output of the rotary-piston engine by approximately 20 to 30 percent. This is of the same order of increase as application to conventional reciprocating-piston engines.

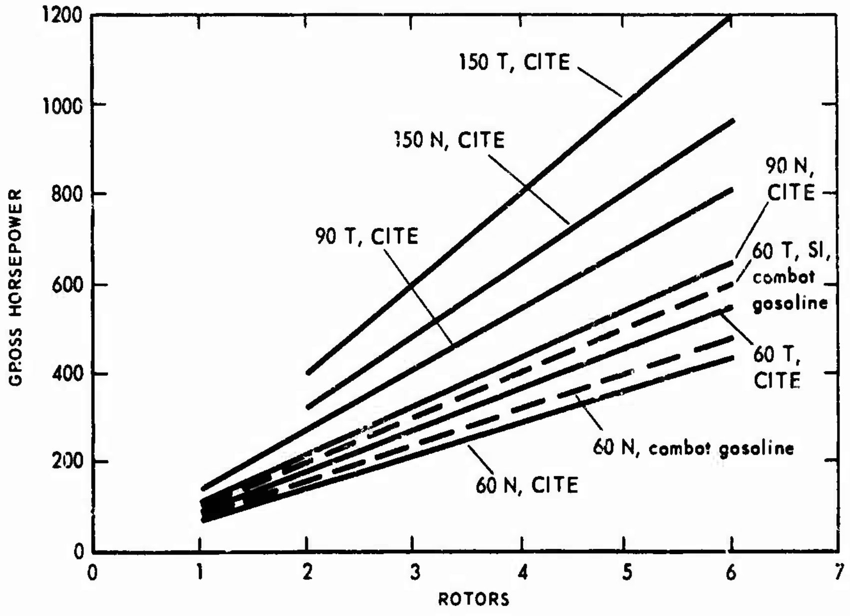


Fig. 1-86—Family Capabilities of Rotary-Piston Engines

All ratings are estimated durability military ratings, present technology. CITE-fuel engines are stratified-charge (hybrid) with spark ignition and fuel injection.
150, 150-in.^{3}-displacement rotor; 90, 90-in.^{3}-displacement rotor; 60, 60-in.^{3}-displacement rotor; N, naturally aspirated; T, turbocharged; SI, spark ignition.

Present automotive rotary-piston engines are liquid-cooled. However, Curtiss-Wright has conducted feasibility and design studies on air-cooling of this engine. Based on these studies they have built and tested an air-cooled single-rotor test engine that demonstrated air-cooling to be as practical as with conventional reciprocating-piston engines. Curtiss-Wright Corporation has developed a twin-rotor air-cooled rotary-piston aircraft engine that incorporates two rotors of 90-in.^{3} displacement. This engine, designated RC2-90-Y, is shown in Figs. 1-87 and 1-88. Burning JP fuel, it develops 310 gross bhp at 6000 rpm and 270 net bhp at 6000 rpm. Minimum specific fuel consumption of this unit is 0.53 lb/hp-hr, utilizing natural aspiration or fuel injection.

Many problems were encountered during development of the rotary-piston engine. Most of these have been solved to an acceptable level, but some still exist. In spark-ignition carbureted induction engines the usual problems of detonation, preignition, rumble, flame quenching, and end-zone knock still exist

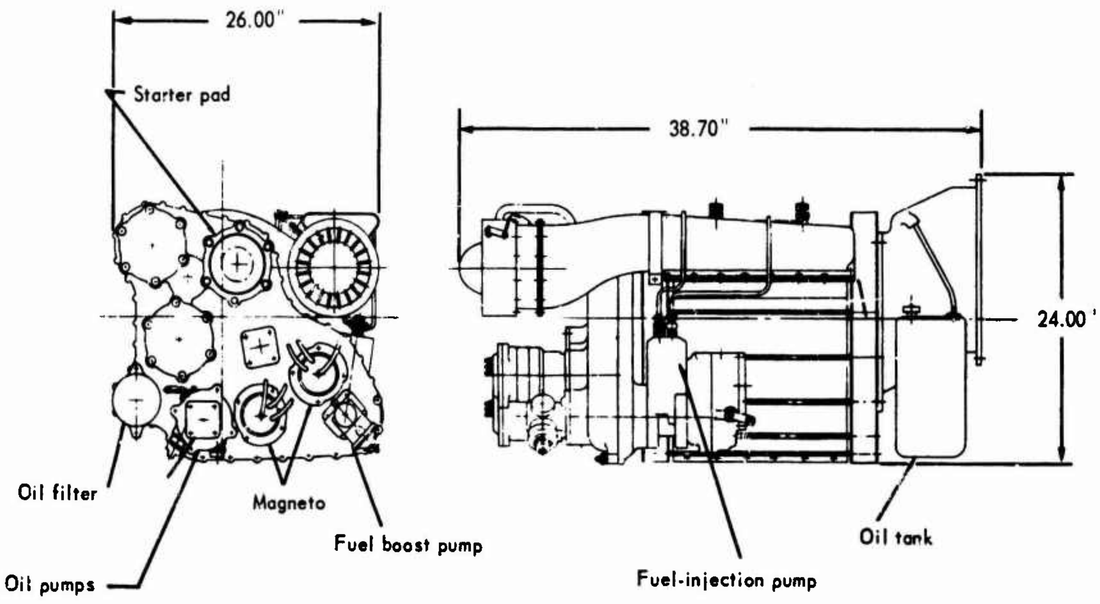


Fig. I-87—Outline Dimensions of RC2-90Y Rotory-Piston Aircroft Engine

BHP gross, 310 at 6000 rpm — Dry weight, 27,816 lb
BHP net, 270 at 6000 rpm — Installed weight, 33,516 lb
Minimum specific fuel consumption, 0.53 lb/hp-hr

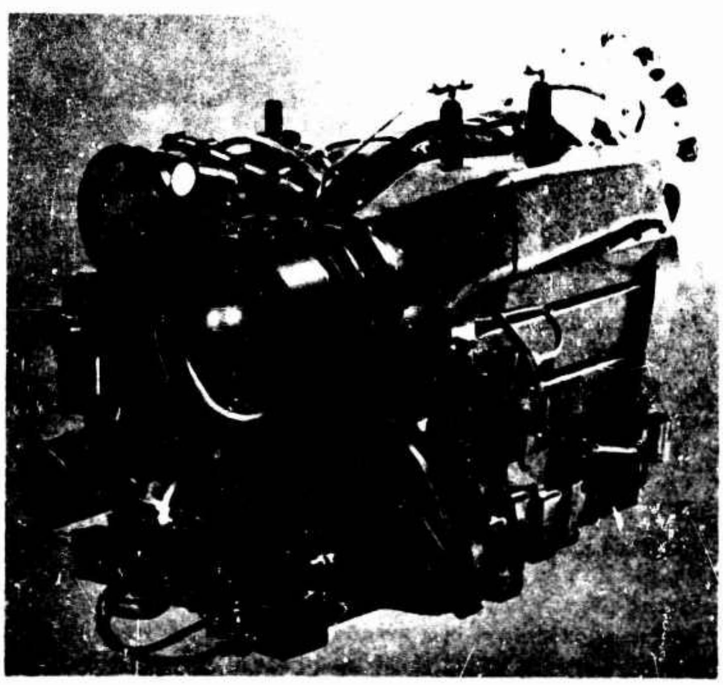
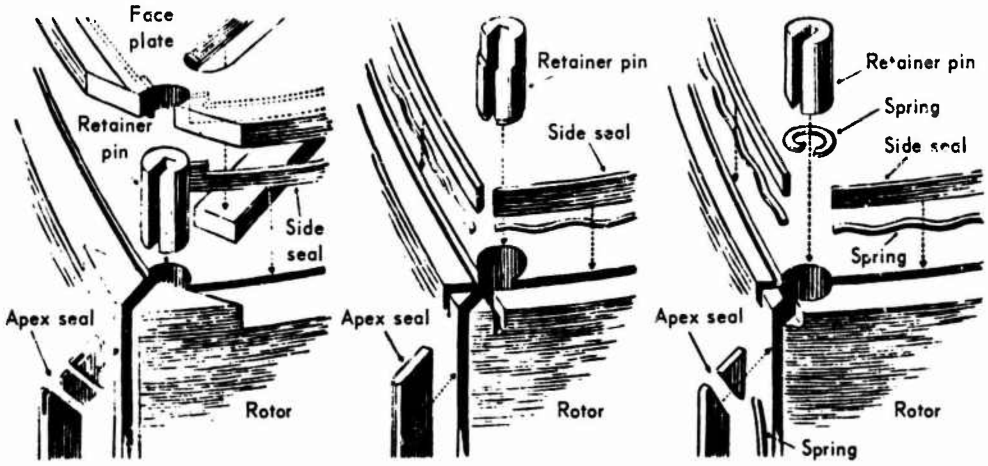


Fig. I-88—RC2-90 Engine, Three-Quarter Rear View

as with conventional engines. The inner-rotor apex and side seals posed many problems during early development of this engine. Figure I-89 illustrates the evolutionary development stages of apex seals and side seals. Early versions of the apex seals (Part a of Fig. I-89) contained many parts and were fragile. The side seals were inserted in grooves in the rotor and on a separate sealing plate. The second-generation apex seals (Part b of Fig. I-89) were single solid bars that exhibited good durability but poor sealing properties at the corners.



a. First-Generation Sealing Configuration (1959)

b. Second-Generation Sealing Configuration (1963)

c. Present-Generation Sealing Configuration (1966)

Fig. I-89—Apex and Side Seal Development of Rotary-Piston Engine

The side seals were contained in grooves machined in the rotor itself. The final sealing configuration (Part c of Fig. I-89) consists of a two-piece bar-type apex seal with a floating triangular section at one end of the element to seal the corners more effectively. This seal design has demonstrated good sealing characteristics and durability. Early rotary-piston engines experienced extreme chatter of the apex seals, which resulted in chatter marks on the epitrochoidal track of the rotor housing. The apex seals are forced against the track by gas pressure, but during gas-pressure reversals the seals tilt in their slots. The seals are also subjected to slight radial movement as the rotor revolves in its trochoidal path. Chopping of the apex seals against the housing track is similar to piston-ring flutter during extreme piston acceleration in conventional reciprocating engines. Curtiss-Wright claims to have eliminated the chatter of the rotor apex seal by coating the housing track with a flame-spray tungsten-carbide metal. A rotor housing from a RC2-60-U5 engine that had been installed in a standard passenger automobile and operated for 15,000 miles at speeds of 60 to 90 mph continuously showed no signs of chatter. The metal-spray coating was checked and 0.00004 in. of wear was measured. Chrome plating may also be used as a coating, although it is less effective than the tungsten-carbide coating. Improvement of the seals is still necessary to minimize combustion-gas and lubricating-oil leakage. Oil consumption of present rotary-piston engines is somewhat greater than that of conventional engines. Oil consumption of the RC2-60-U5 engine installed in a passenger automobile is approximately 1 qt/1000 miles compared with approximately 1 qt/1300 miles for a conventional piston engine.

Cooling of the rotary-piston engine has been a problem in the past. Combustion is confined to the same area of the inner rotor for each combustion cycle, causing a hot spot. This problem has evidently been solved through redesign of passages for proper coolant distribution. Total heat transfer to the water jacket and oil of the rotary-piston engine is comparable to that of current reciprocating engines, indicating that a cooling system of the same size and weight will be required.

Fuel consumption of rotary-piston spark-ignition engines is comparable to that of reciprocating-piston spark-ignition engines. The RC2-60-U5 rotary-piston engine (with spark ignition and carbureted fuel induction) has demonstrated a best-point minimum specific fuel consumption of 0.53 lb/hp-hr at part-throttle and 0.59 lb/hp-hr at full throttle. This performance is comparable to that of a conventional reciprocating-piston engine of similar power output. Figure I-90 illustrates the automotive performance characteristics of the RC2-60-U5 engine. Figure I-91 shows a comparison of the performance characteristics of a rotary-piston engine with those of a conventional reciprocating engine of similar power output.

To achieve lower fuel consumption the rotary-piston engine would have to operate on the diesel cycle or incorporate a hybrid combustion system.

It does not appear that the rotary-piston engine could be successfully adapted to the diesel cycle for high-output engines owing to the high compression pressures required. The rotor-sealing problems would be compounded. Some European licensees are pursuing the adaptation of the rotary-piston engine to the diesel cycle with low-output engines that would be much heavier and bulkier than spark-ignition units. The Curtiss-Wright Corporation believes that a hybrid combustion system incorporating spark ignition and a stratified-charge fuel-injection system is a better approach than the diesel cycle for reduced fuel consumption. The hybrid combustion system would also have the advantage of a multifuel capability. Curtiss-Wright is currently engaged in limited development work on this combustion system.[4] Figure I-92 illustrates the achievements to date on a fuel-injected single 60-in.^{3}-displacement rotary-piston engine.

The Curtiss-Wright rotary-piston spark-ignition gasoline engines, under moderate loading, are producing 0.77 hp/lb (1.3 lb/hp) of engine weight, 33 hp/ft^3 of engine volume at automotive power ratings. Based on an estimated military durability rating, the power output is 0.66 hp/lb (1.5 lb/hp) of engine weight and 29 hp/ft^3 of engine volume. This compares favorably with a conventional spark-ignition gasoline engine whose specific weight at military rating is 3.4 lb/hp and whose specific output is 15.7 hp/ft^3 of volume. Figure I-93 illustrates a size and weight comparison of the rotary-piston engine and a contemporary V-8 automotive engine currently in use in military tactical vehicles. The rotary-piston engine is 56 percent lighter and 56 percent smaller than the V-8 piston engine. Figure I-94 illustrates the general arrangement of the RC2-60-U5 engine.

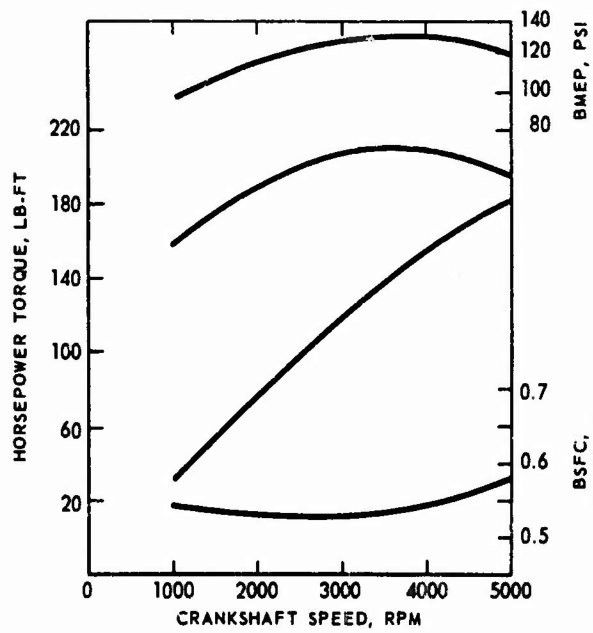


Fig. I-90—Performance Characteristics of R C2-60-U5 Rotary-Piston Engine

(185 hp, twin rotor, 120-in.^{3} displacement) (60°F, 29.92-in. Hg)

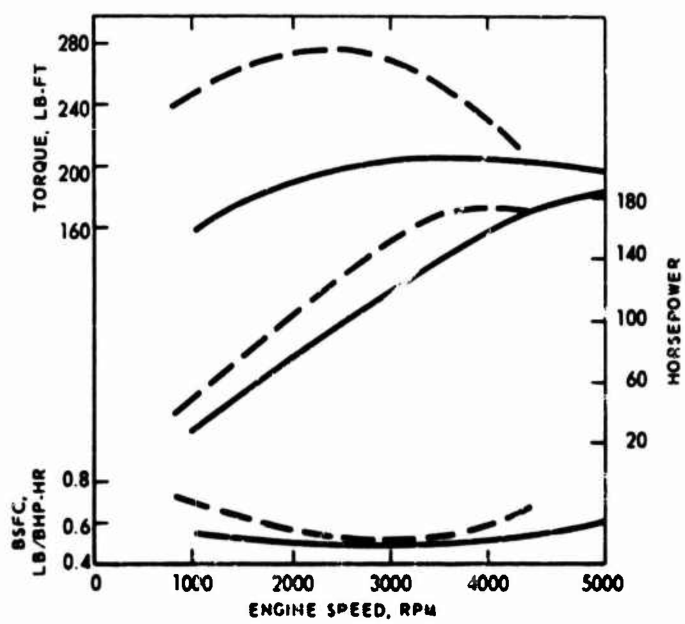


Fig. I-91—Comparison of Performance Characteristics of a Rotary-Piston Engine with Those of a Conventional Reciprocating-Piston Engine

——— Rotary-piston engine
— — Conventional reciprocating-piston engine

The rotary-piston engine is exceedingly interesting in that it is comprised of so few simple parts that it appears to adapt itself to a throw-away concept. The rotary engine could be designed so that on malfunction or failure of the basic engine assembly containing the power-producing components it could be discarded rather than repaired or overhauled. The accessory components (such as carburetor, distributor, fuel pump, starter, generator, etc.) could be salvaged from the damaged unit and placed on a new basic-engine assembly. This concept could result in greatly reduced maintenance requirements.

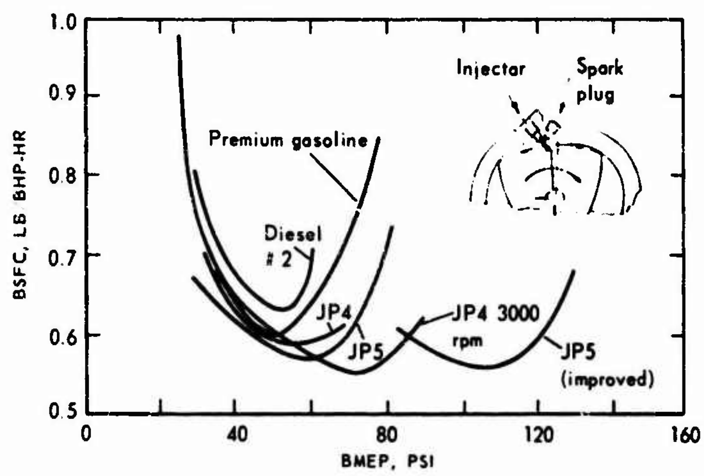


Fig. I-92—Fuel-Injection Performance of Rotary-Piston Engine

(Single rotor, 60-in.^{3} displacement)
5000 rpm except as noted, 8.5:1 CR, RC1-60.

Manufacture of the rotary-piston engine would require special tooling for machining and grinding of the rotor, rotor housing, and side housings. The remainder of the engine could be produced with little or no modification of existing automotive tooling.

Development work by the Curtiss-Wright Corporation on the rotary-piston engine has at time of writing been accomplished with only corporate funds.

Conclusions

Projected development programs of the Curtiss-Wright Corporation to improve and increase the potential of the rotary-piston engine are:

(a) A multifuel capability
(b) An improved combustion system
(c) A hybrid combustion system
(d) An improved fuel-injection system
(e) Turbosupercharging
(f) Increased bmep
(g) Air cooling
(h) Improved durability

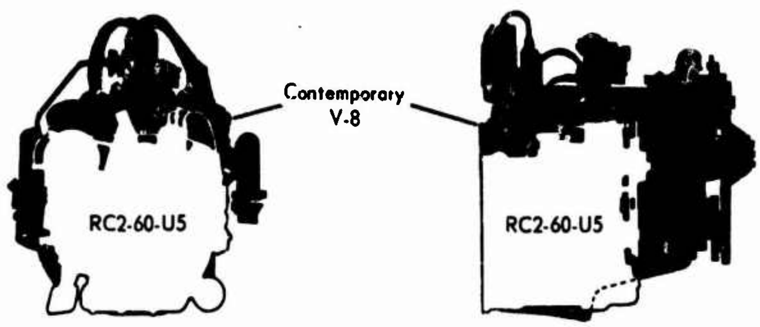


Fig. I-93—Comparison of Rotory-Piston Engine with Contemporary V-8 Engine

Characteristics	Rotary-piston engine	Contemporary V-8[a] engine
Horsepower	160[b]	160
Weight, lb	242	550
L × W × H, in.	18.0 × 22.1 × 21.5	26.1 × 26.2 × 31.5
Volume, ft^3	5.5	12.5

[a]Militarized commercial engine.
[b]Estimated military durability rating.

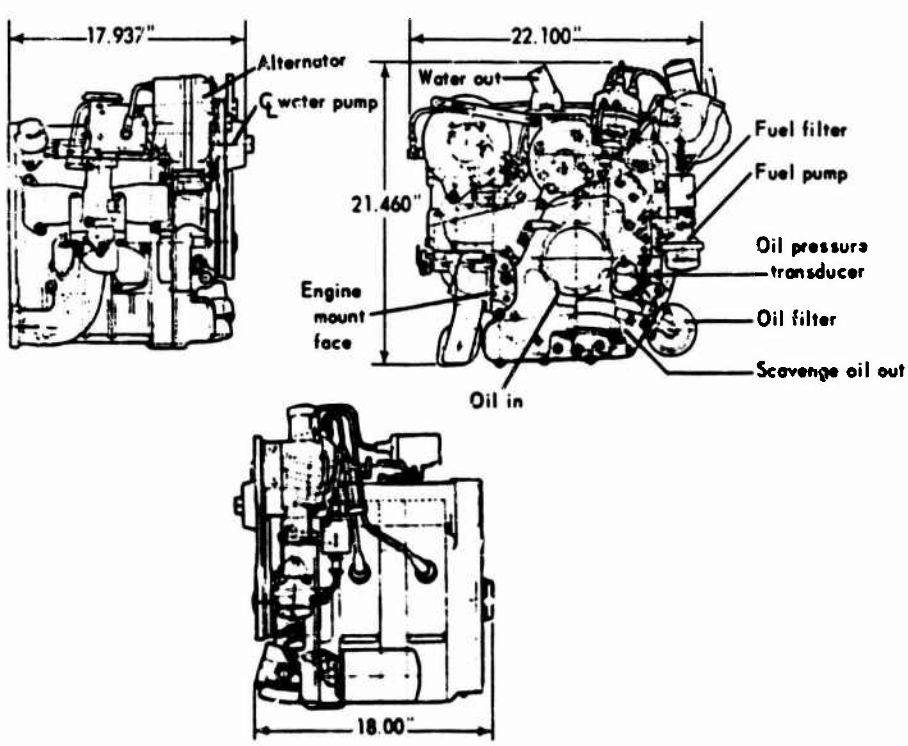


Fig. I-94—General Arrangement of RC2-60-U5 Rotary-Piston Engine

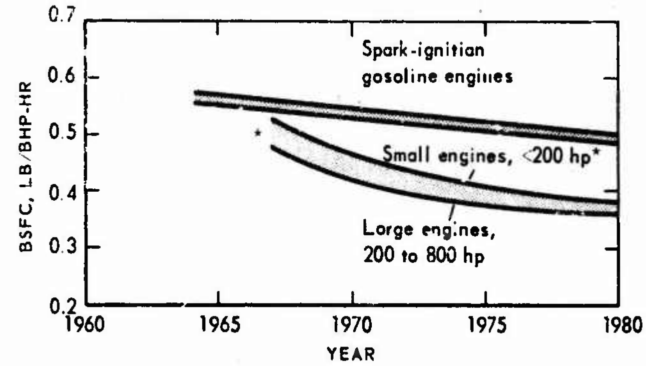


Fig. I-95—Trend Forecast of Fuel Consumption for Rotory-Piston Engines

*Engines incorporating fuel injection and hybrid combustion system, operating on heavy fuels (diesel, CITE, JP5, etc.).

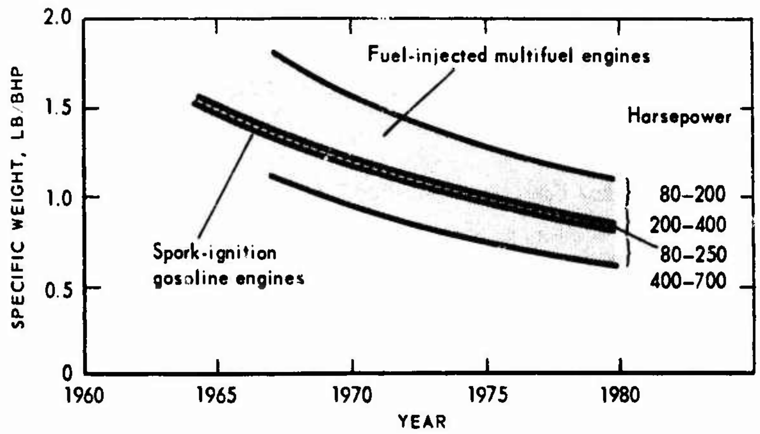


Fig. I-96—Trend Forecast of Specific Weight for Rotory-Piston Engines

All (except spark-ignition gasoline) engines incorporate fuel injection, and future capabilities are based on eventual incorporation of turbocharging and a hybrid combustion system. Forecast for spark-ignition gasoline engines is based on eventual incorporation of turbocharging.

The application of some of the desired improvements (such as items a, c, d, e, and f) would be primarily for military use. Military funding would be required to ensure achievement of the stated improvements within a reasonable time. The successful completion of the projected programs would provide the military with engine improvements such as:

(a) Reduced fuel consumption
(b) Multifuel capability
(c) Increased power output at reduced weight
(d) Increased power output for engine size (volume)
(e) Reduced maintenance requirements
(f) Lower production costs than present power sources

The trend-forecast charts (Figs. I-95 to I-97) illustrate the estimated level of technological achievement with regard to the rotary-piston engine through 1980.

Cooperative Government support of the Curtiss-Wright Corporation's R&D programs could produce a rotary-piston engine that could improve the physical and performance characteristics of many future tactical vehicles.

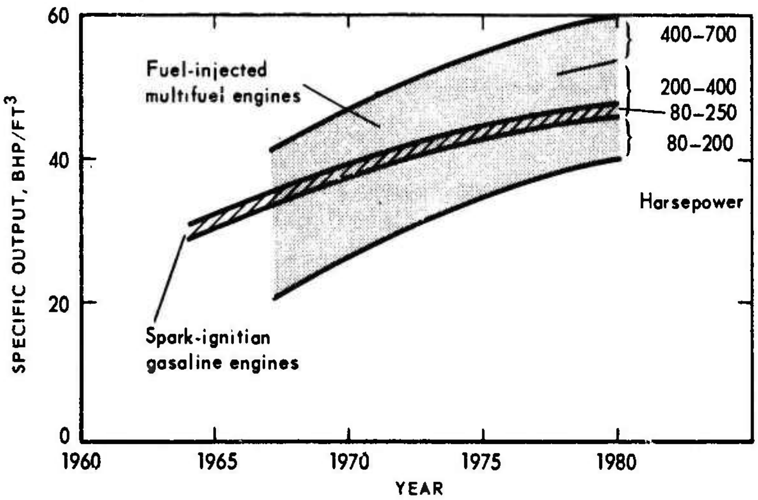


Fig. I-97—Trend Forecast of Specific Output for Rotary-Piston Engines

All (except spark-ignition gasoline) engines incorporate fuel injection, and future capabilities are based on eventual incorporation of turbocharging and a hybrid combustion system. Forecast for spark-ignition gasoline engines is based on eventual incorporation of turbocharging.

RENAULT-RAMBLER ROTARY-PISTON ENGINE

There have been many other attempts throughout the world to develop a rotary-piston engine. The majority of rotary-engine designs are impractical for one reason or another. Some are too costly to manufacture, some have poor combustion-chamber shapes, and some are kinematically as complex as the reciprocating engine.

Probably the most notable attempt to develop a rotary-piston engine on other than the Wankel or Curtiss-Wright Corporation principle is the Renault (French automaker) and American Motors Corporation (Rambler) concept. The Renault-Rambler rotary-piston engine concept is shown in Fig. I-98. It differs from the Wankel concept in that the combustion chambers are stationary, and one lobe of the rotor acts as a piston much like that in a reciprocating-piston engine.[5] The gases are not transported or transferred within the outer housing. The Renault-Rambler rotary-piston engine is comprised of five combustion chambers arranged radially about a central crankshaft and contained in a single-unit housing. The rotor has four lobes supported by a set of internal and external gears that coordinates rotor rotation and crank rotation so that the rotor does not touch the outer housing, and clearance is always maintained between the rotor and the housing. The circulating motion of the rotor has the same effect the conventional engine's reciprocating piston. Conventional poppet valves actuated by valve mechanisms are utilized for intake and exhaust functions. The rotor seals are supported in the outer housing and are stationary. The seals are water-cooled from the water jacket in the outer housing. (This is in contrast to the Wankel engine, in that the seals are mounted in the rotor and rotate with it, and are oil-cooled.) The Renault-Rambler engine utilizes spark ignition and carbureted fuel induction and operates on the 4-stroke Otto-cycle principle. The Renault Company estimates that this engine will eventually develop 3 hp/lb of engine weight.[6] They are experimenting with lightweight materials, including high-temperature plastics, to reduce engine weight and costs.

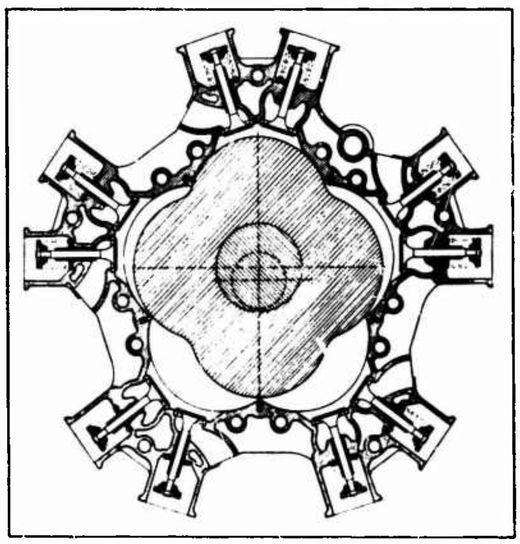


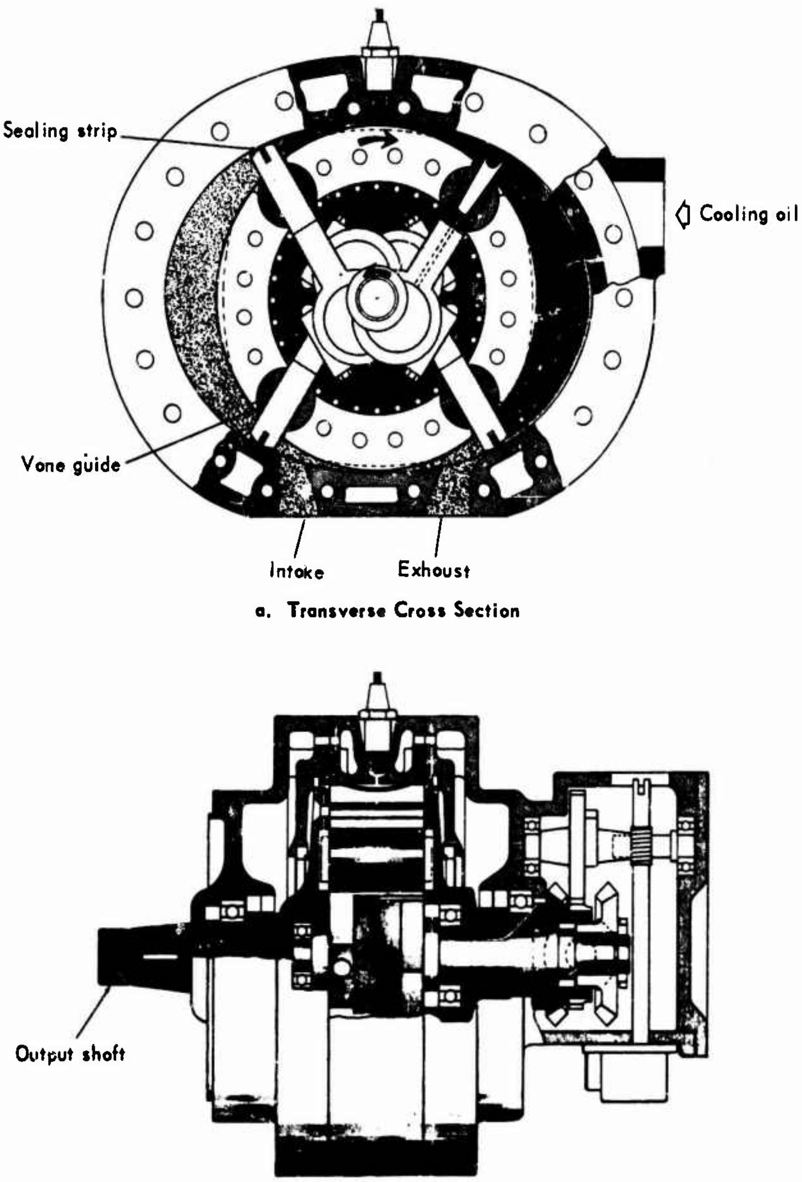
Fig. I-98—Cross Section of Renault-Rambler Rotary-Piston Engine

Although the Renault-Rambler rotary-piston engine offers much the same simplicity of power-producing components as the Wankel engine, it has disadvantages in design arrangement that conflict with the initial purpose. The valve arrangement requires complex valve-train mechanism and intake and exhaust manifolding.

The Renault-Rambler rotary-piston engine does not appear to offer any advantages over other power sources and therefore does not warrant development by the Government.

BALVE ROTARY ENGINE

The Balve engine, whose patent rights are assigned to Etudes et Recherches Techniques "Bafelux," Luxembourg, is the result of an attempt to develop a rotary engine whose rotor and housing are simpler to manufacture than the



a. Transverse Cross Section

b. Axial Cross Section

Fig. 1-99—Cross Section of Balve Rotary Engine

rotary-piston engines previously discussed. The Balve engine operates on the conventional 4-stroke Otto-cycle principle, utilizing spark ignition and either carbureted fuel induction or a fuel-injection system. The engine uses ported intake and exhaust openings and therefore contains no valves or mechanisms associated with them. The basic concept of the Balve engine is a drum with reciprocating vanes attached to a crankshaft that operates within an elliptical track in the outer housing. This is in contrast to the Wankel-engine concept of a trochoidal rotor operating eccentrically within an epitrochoidal track. Figure 1-99 illustrates the transverse and axial cross sections of a Balve engine and reveals how the conventional 4-stroke combustion process results from the cyclic variation of the working space.[7] The air-fuel mixture is aspirated at the inlet port and compressed, ignited, burned, expanded, and exhausted. Each of the four working spaces between vanes exerts one power stroke for each rotor revolution. All cycles occur within the same section of the engine, which results in a hot spot. This can be controlled by effective

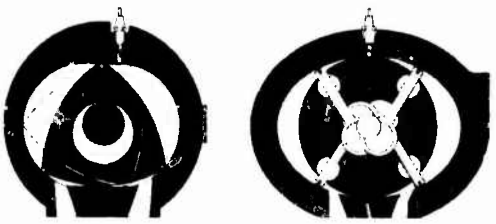


Fig. I-100—Comparison of Wankel and Balve Engine Geometry

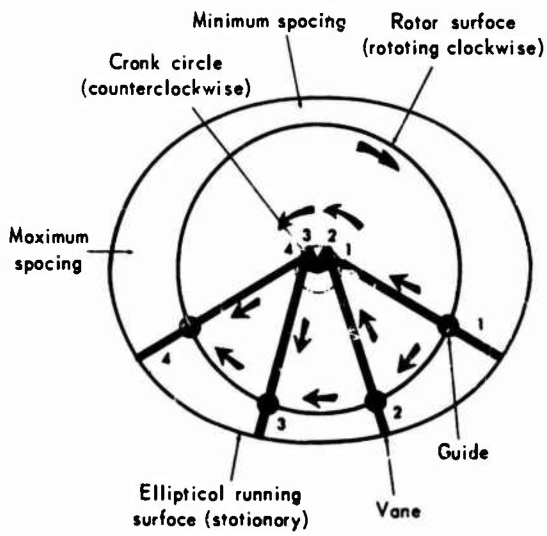


Fig. I-101—Basic Kinematics of Balve Engine

coolant flow and coolant passage design. The cross section of the engine illustrates the arrangement of the outer housing, rotor assembly, vane crankshaft, reciprocating vanes, seals, and cylindrical vane guides. The rotor is supported by antifriction bearings set in the outer housings, and the vane crankshaft is supported within the rotor (and attached to the rotor) by antifriction bearings. The vanes are journaled to the crankshaft eccentrics, which are in four separate transverse planes. The vane ends carry the small seal strips that separate the four working spaces. Annular rings (similar to piston rings) seal the rotor and housing both axially and radially. These rings are cooled and lubricated by the internal recirculating oil.

Figure I-100 compares the geometry of the Wankel and Balve engines.

The basic kinematics of the Balve engine are shown in Fig. I-101. As the rotor drum rotates clockwise, the crankshaft turns counterclockwise. The crank and rotor have identical rotating speeds. The vane, hinged at the lower end to a crank arm, is free to slide and pivot within its guide. The vane tip therefore describes an approximately elliptical path, shown by a single vane in the successive positions 1-2-3-4. The volume enclosed between two successive vane ends, the circular rotor, and the elliptical running surface of the housing, therefore, varies cyclically in magnitude. These variations reach maxima and minima twice per revolution.

The basic theory of this engine concept is that an elliptical track and circular rotor (as compared to the epitrochoidal track and trochoidal rotor of the Wankel unit) would be less difficult to manufacture and would therefore be less costly. Also, service life of sealing strips would be greater due to the simpler geometry of the track. Although the above advantages of the Balve engine may be generally correct, the concept has not eliminated the complexity of reciprocating parts and assemblies, guides, and other elements common to the conventional reciprocating-piston engine. These disadvantages conflict with the initial purpose of developing a simple power source.

It does not appear that the Balve rotary engine offers any advantages over conventional engines or other rotary-piston engines that warrant development by the military.

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Chapter 6

STEAM ENGINES

The first source of mechanical power used in vehicular applications was the steam engine. The heat energy utilized to produce steam was obtained from burning wood, coal, or hydrocarbons.

The basic types of steam engine are the reciprocating, the turbine, and the combination of these two. Each of these three types can be operated as a noncondensing system (which expends exhaust steam into the atmosphere) or as a vapor cycle (which condenses exhaust steam and reuses it in the system). The noncondensing system requires the constant addition of purified water to replace the water expended. The vapor-cycle engine requires a heat exchanger and a cooling fan to condense the steam after it passes through the exhaust ports. Although replenishment of water is not required, the heat exchanger and cooling fan make the vapor-cycle system larger and heavier than the noncondensing engine (see Figs. I-102 and I-103).[1]

When internal-combustion engines and hydrocarbon fuels became readily available at a reasonable cost, the steam engine was replaced by the reciprocating internal-combustion engine. The railroad locomotive was the last holdout for steam-powered engines. However, these steam engines were eventually replaced with internal-combustion diesel engines after industry developed the diesel engine to meet the horsepower, size, speed, and torque requirements of the railroad locomotive.

DISCUSSION

Interest in vehicles utilizing the steam engine is revived every few years, in particular with regard to those vehicles incorporating the vapor-cycle system. In 1949 the Yuba Manufacturing Company of Benicia, California, announced the development of a prototype steam tractor, for the civilian commercial market, that incorporated four reciprocating steam engines, one for each wheel. The power plant, a vapor-cycle system, had a common steam generator for the four engines. Each engine had four cylinders, horizontally opposed, and produced 50 hp. The engines were mounted vertically to an angle drive that drove the planetary axle end. A heat exchanger and a cooling fan were incorporated to condense the exhaust vapors. The system was operated for approximately 10 years. Although firm records do not exist to show the frequency or extent

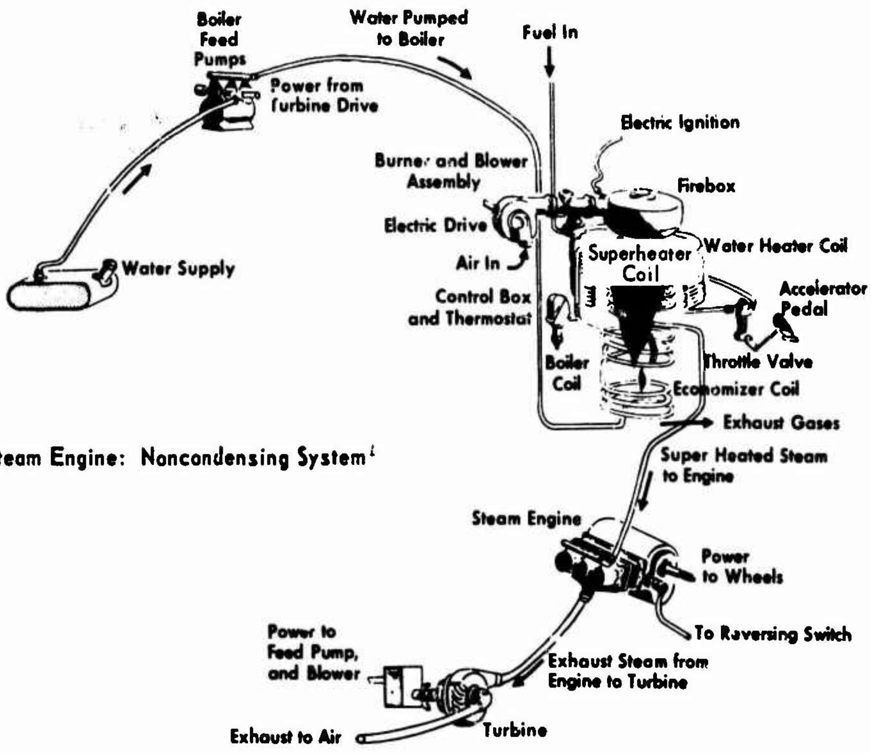

Fig. I-102—Steam Engine: Noncondensing System[1]

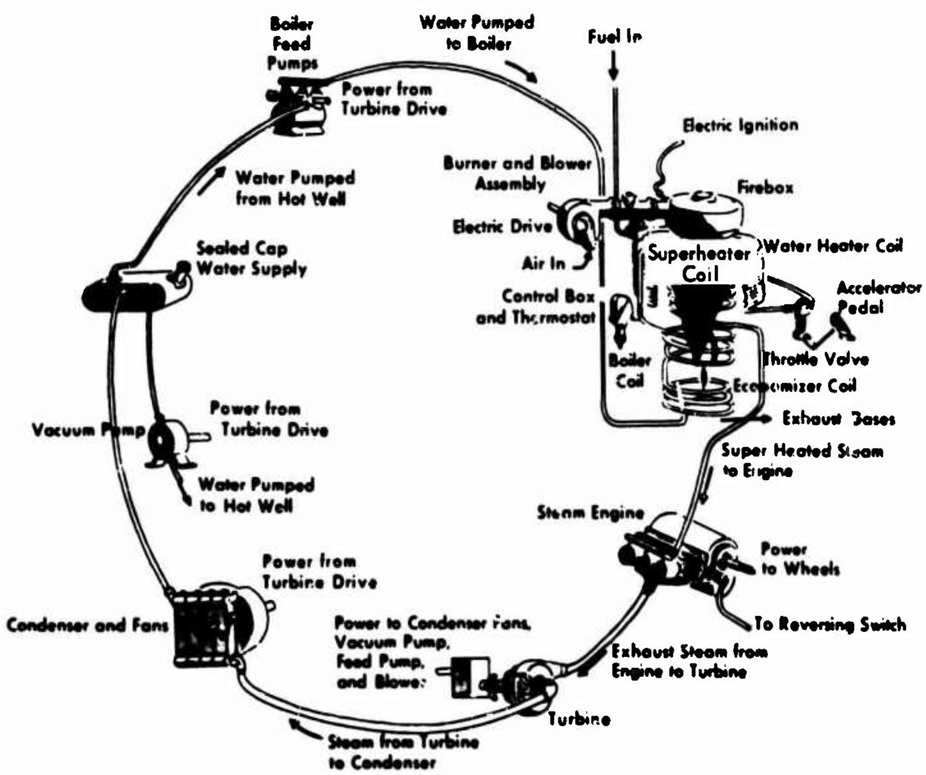

Fig. I-103—Steam Engine: Closed Vapor Cycle System[1] of maintenance required for this system, miscellaneous information received indicated minimum maintenance requirements.

Personnel at the Yuba Manufacturing Company stated that the steam power system is heavier and bulkier than an internal-combustion reciprocating engine offering comparable performance features. The inference was made that it would be difficult to reduce the size and weight of the steam power system. However, the fact that Yuba was not concerned about the additional weight and size of the steam tractor was made evident during the discussion. Statements were made that the tractor was large enough to handle the engine size and that the system did provide additional traction for increasing tractor drawbar pull. The system could generate sufficient pressure in 30 to 40 sec to propel the tractor. A market survey made by Yuba marketing personnel resulted in the decision to discontinue the steam-tractor program.

In 1957 the Operations Research Office of Johns Hopkins University and the Chrysler Corporation prepared a joint mobility study report that presented the advantages of a vapor-cycle power system for a 10-wheeled vehicle.[2] (The concept was similar to that employed in the development of the prototype steam tractor produced by the Yuba Manufacturing Company.) The estimate that the engine would produce 25 hp per wheel was made, yielding a total of 250 hp for the 25-ton vehicle envisioned. A weight of 4540 lb was estimated for the complete system.

The vehicle, as proposed, was expected to operate for 2000 miles before breakdown. The routine maintenance of the system would be less than that required by present-day internal-combustion systems because the vapor-cycle system has fewer moving parts. An analysis of the system indicated that by current standards the vehicle appears to be underpowered, too large, and too heavy, considering the total horsepower output (see Fig. I-104). Specific reasons why this vehicle was not developed were not determined by the study group, but from the data obtained it can be assumed that the proposed steam-powered system did not provide sufficient advantages over comparable and available internal-combustion systems to justify further development.

In July 1962 the Convair Division of General Dynamics Corporation completed a study[3] made to determine the feasibility of applying principles of a vapor-cycle steam-turbine system to any tactical vehicle of the US Army. The results of the study indicated that it was feasible to design and develop a 500- to 600-hp steam-turbine power system for a battle tank. The results of the study were presented to the Army, and a design for the battle tank proposed.

Two turbines, each rated at 250 to 300 hp and driving independently, were proposed for the battle tank to eliminate the need and complexity of a steering-transmission unit. The overload capability of the turbines allowed for an emergency power output of from 50 to 100 percent above rated capacity. This power system was designed to have a multifuel capability, low noise level, and high reliability. Calculations indicated that this steam-power system would have an overall efficiency of 27 percent and that the turbines would furnish 880 lb-ft of torque at stall. Sufficient steam pressure to propel the vehicle could be obtained in 30 to 40 sec. At maximum battle-tank speed 440 lb-ft of torque would be furnished.

An estimate of 4000 hr of tank operation before major engine overhaul was given. The maintenance requirements of the power system would be less than those of an internal-combustion system owing to fewer moving parts.

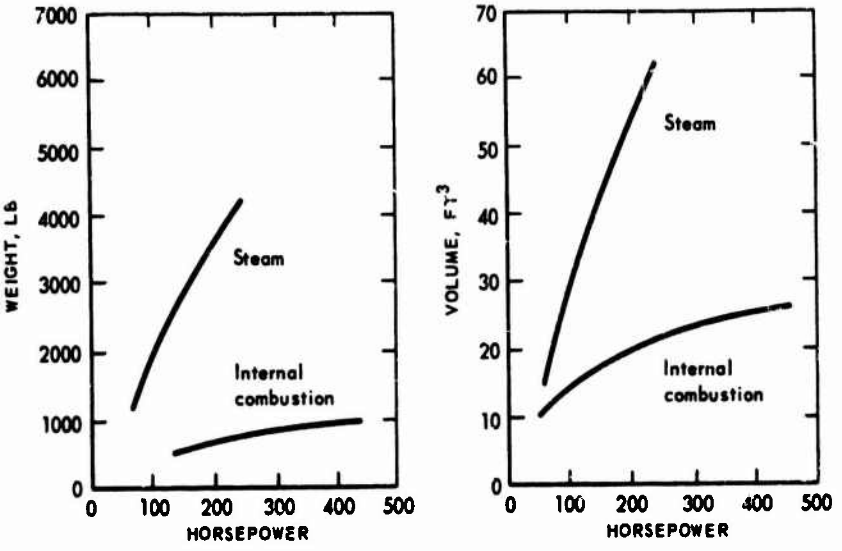

Fig. I-104—Physical Comparison of Steam Engine with Internal-Combustion Engine

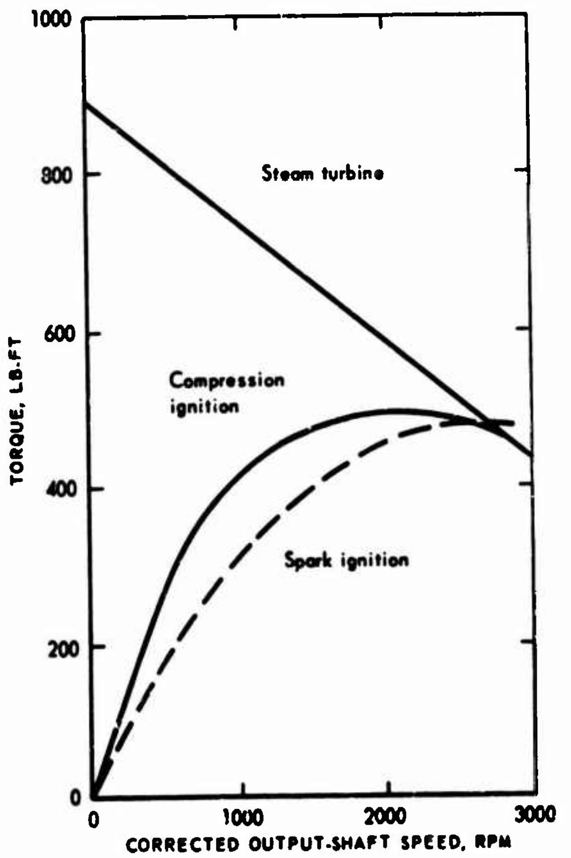

Fig. I-105—Torque Comparison of a 250-hp Steam Turbine with Two Typical Internal-Combustion Engines

A favorable feature of the steam-turbine engine is its excellent torque output at low engine speeds when compared with that of an internal-combustion engine operated at the same speed (see Fig. I-105). Although the steam-engine power system would be improved with the development of better and lighter materials, and the overall weight and size would decrease, these improvements are predicted to be marginal.

CONCLUSIONS

One advantageous characteristic of the steam engine, unsurpassed by most other types of engine, is its quiet operation. This characteristic assumes major importance in those situations where vehicles must operate in silence. In addition, steam-power systems may be operated with various types of fuel and therefore can comply with the present fuel policy of the military.[2,3,4]

The disadvantages of a steam engine for a tactical vehicle, as compared to an internal-combustion engine, are given in the following paragraphs.

Weight. Steam engines are heavier than gasoline or diesel engines of equivalent horsepower. Concepts have not been proposed that would reduce the weight of steam-power systems in the foreseeable future.

Size. The steam engine is considerably larger than a present internal-combustion engine. A noncondensing system would require a large water reservoir. A closed vapor-cycle system would require a heat exchanger and a cooling fan.

Efficiency. The fuel efficiency of a steam engine, at optimum three-quarter throttle, is approximately 27 percent. The fuel efficiency of an internal-combustion gasoline engine is approximately 35 percent, and that of an internal-combustion diesel engine is approximately 38 percent.

Warm-Up Time. Tactical vehicles that use steam engines require start-up time to generate a head of steam sufficient to permit any high-level power output in excess of that required to propel the vehicle at a limited speed.

Maintenance. Steam-engine maintenance time may be regarded as minimal when compared with present system requirements. A strong disadvantage is that personnel would have to be indoctrinated to maintain the vapor-cycle system, thereby necessitating an increase in maintenance training.

Spare Parts. Additional spare parts would be required in the military supply system for vehicles having a steam engine since the spare parts for these engines are not common to internal-combustion engines.

The RAC analysis of various steam-engine concepts and prototype installations indicated that steam engines for tactical vehicles do not offer any significant advantages over present internal-combustion engines. Therefore R&D programs for steam engines applicable to tactical vehicles are not warranted unless the development of vehicles offering silent operation is considered more important than overall volume and weight limitations.

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Chapter 7

DYNASTAR ENGINES

The Dynastar engine is an unconventional 2-stroke-cycle engine designed to provide uniflow scavenging and charging with unsymmetrical port timing* by means of dual-piston U-shaped cylinders interconnected in a unique arrangement. The Dynastar engine is neither a radial nor an X-configuration engine but can be considered a folded-up version of an opposed-piston engine in a lighter and more compact package. The engine concept is based on the theory that two pistons are lighter than one geometrically similar unit of the same total piston area, and two small crankshafts are lighter than one geometrically similar crankshaft having the same operating parameters. The folded-up design configuration permits better utilization of space and results in the lighter and more compact engine package. This arrangement also permits a reduction in crankcase size and weight. Either spark ignition or compression ignition and air or liquid cooling are applicable to the Dynastar engine.

Design and development work on the Dynastar engine concept was initiated in 1958 by the Thiokol Chemical Corporation, which obtained exclusive license rights from the holder of the Dynastar patents.

The basic Dynastar design,[1] shown in Fig. I-106, consists of four U-shaped paired cylinders positioned in a common plane. The paired cylinders are oriented to each other at an angle selected to give a desirable geometrically phased† lead to exhaust-port opening in relation to inlet-port opening.

The paired pistons of each U-shaped cylinder pair are connected to a true-motion octagonal plate or frame by individual connecting rods. The frame links the crankshafts so that they rotate in phase. Any point on the frame inscribes a true circle having a radius equal to the crank throw. The connecting rods are linked to the frame with conventional knuckle pins. Each knuckle pin is journaled in the frame and travels in a true circle of crank-throw radius. This action imparts identical motion to all pistons when the connecting rods are the same length and the orbital center of each knuckle pin is on the center line of its respective cylinder.

Perfect dynamic balance[2] can be achieved by counterweighting the crankshafts. The two crankshafts are connected by gearing, as shown in Fig. I-107.

*"Unsymmetrical port timing" means that exhaust ports open in advance of inlet ports but also close before inlet ports.

†"Geometrically-phased" is defined as "the phase relation of events established by the geometry of the engine, exclusive of valve-port size or location."

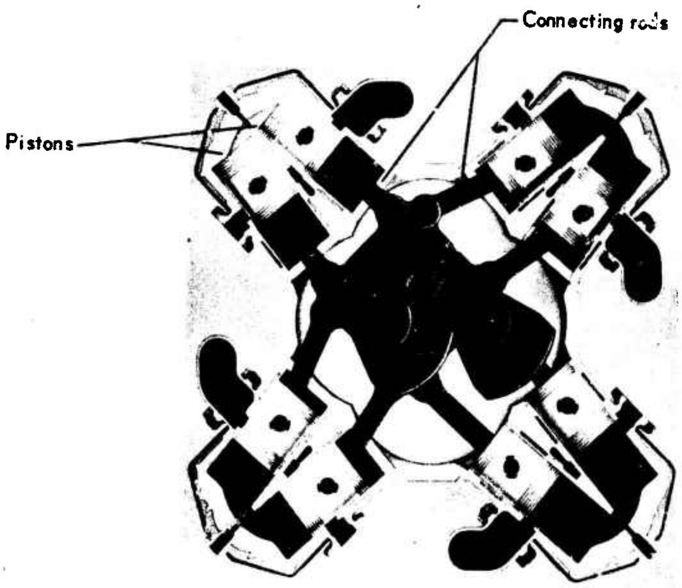


Fig. I-106—Cross Section of Dynastar Engine

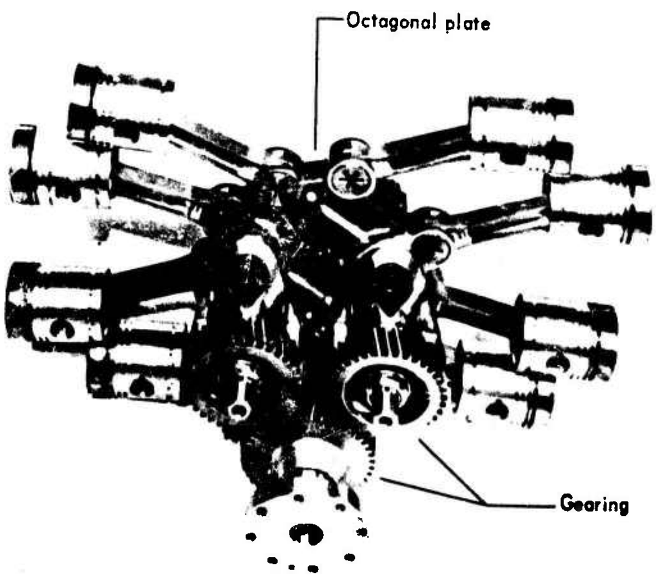


Fig. I-107—Power-Producing Components and Geared Offset Drive-Shaft Configuration of Dynastar Engine

This gearing provides a synchronous motion to the frame when the crank throws are in exactly the same plane. The synchronizing gear also provides a single common output shaft and has the additional capability of providing a wide selection of output-shaft speeds and positions.

DISCUSSION

The configuration of present Dynastar spark-ignition and compression-ignition developmental engines consists of four cylinder blocks symmetrically disposed in one plane about a common central axis. This configuration is shown in Parts a and b of Fig. I-108. However, the engine may also be constructed as a two-row (see Part c of Fig. I-108), three-row, or four-row unit. The family capabilities of the Dynastar engine are shown in Table I-13. The stacking of more than three or four rows of cylinders appears impractical owing to the long crankshaft required.

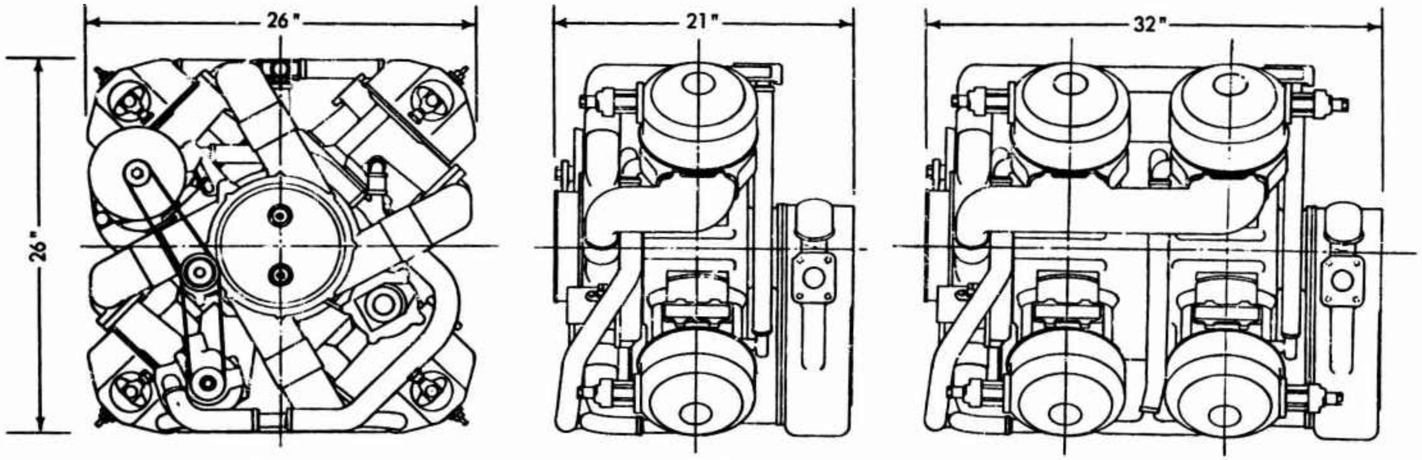
TABLE I-13

Family Capabilities of Dynastar Compression-Ignition Engines

Configuration (3-in.-bore cylinders)	Horsepower	Configuration (5.25-in.-bore cylinders)	Horsepower
Single row	100	Single row	350
Single row, turbocharged	135	Single row, turbocharged	450
Double row	200	Double row	700
Double row, turbocharged	270	Double row, turbocharged	1000
Triple row	300	Triple row	1050
Triple row, turbocharged	405	Triple row, turbocharged	1500

Present Dynastar engines are liquid-cooled. However, design studies conducted by the engine developer have shown air cooling of these units to be as practical as the air cooling of present conventional reciprocating-piston engines. Both spark- and compression-ignition Dynastar engines use a centrifugal blower for cylinder scavenging and charging. These engines can be designed with cylinder banks and output shaft positioned either horizontally or vertically (see Fig. I-109). The engines can be manufactured using present automotive-engine tooling, for the most part, since the power-producing components are similar to the reciprocating components of a standard automotive engine.

A meeting with Thiokol Chemical Corporation representatives disclosed that the Dynastar power plant is considered as being in the early stages of development. The fuel consumption of the present developmental engines is higher than that of present conventional and competitive compression-ignition engines. Current Dynastar compression-ignition engines incorporate a prechamber combustion system. However, the manufacturer expressed the belief that development of both an open-chamber combustion system and an improved fuel-injection system will reduce Dynastar fuel consumption and increase power output.



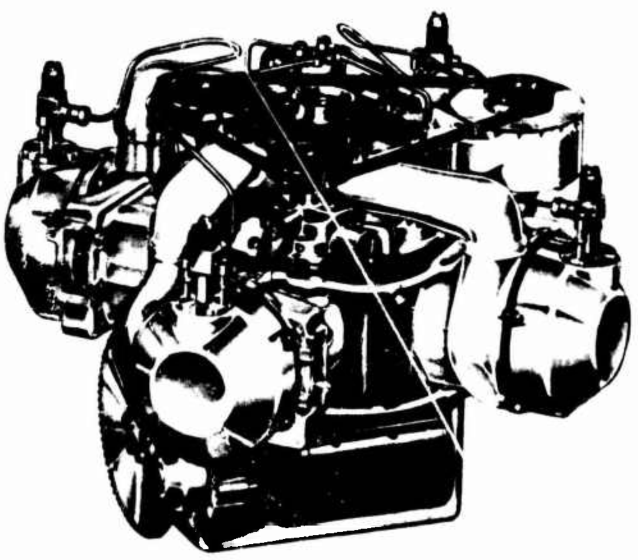
a. Front View

b. Side View, Single Row
100 hp, 8 cyl

c. Side View, Double Row
200 hp, 16 cyl

Fig. I-108.--Dynastar Compression-Ignition Engine Configurations

a. Vertical Cylinder Bank—Horizontal Output Shaft



b. Horizontal Cylinder Bank—Horizontal Output Shaft

Fig. I-109—Dynastar Engine Configurations

Dynastar spark-ignition engines under moderate loading are producing 1 hp/lb of engine weight, 41 hp/ft^3 of engine volume, and at aircraft power loading 2 hp/lb of engine weight.[3] The present Dynastar compression-ignition engines are producing 0.25 hp/lb of engine weight and 12.2 hp/ft^3 of engine volume at moderate loading. A 100-hp compression-ignition engine is at present undergoing US Coast Guard tests in a 19-ft PICWAN-type boat. To date this engine has demonstrated satisfactory performance.

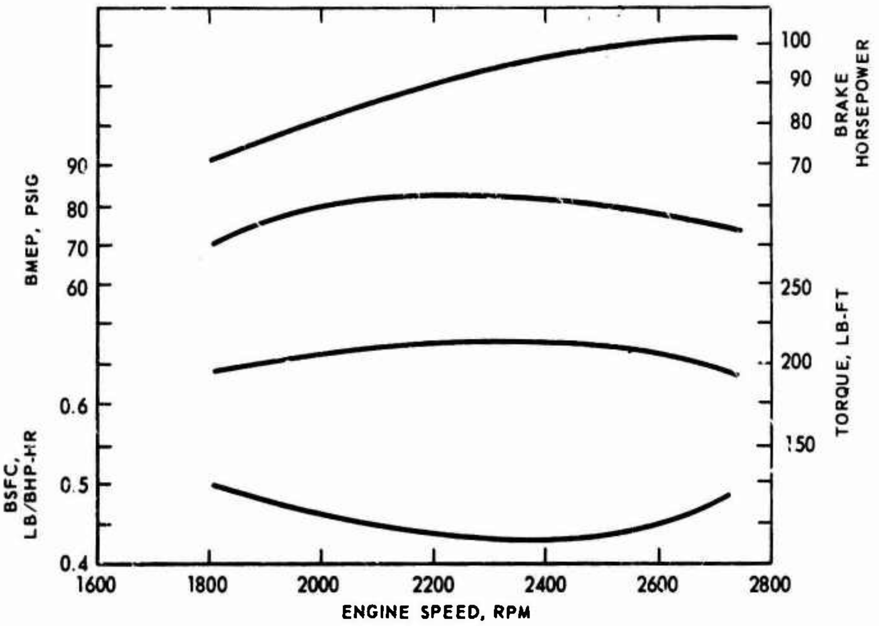


Fig. I-110—Performance Characteristics of the Dynastar Compression-Ignition Engine

(100 hp, 8 cylinders, 3-in. bore, 3.5-in. stroke)
(60°F, 29.92-in. hg)

Figure I-110 illustrates the performance characteristics of the 100-hp 8-cylinder Dynastar compression-ignition engine. The fuel consumption curve shows a best-point specific fuel consumption of 0.44 lb/bhp-hr. Many conventional compression-ignition engines in this horsepower range have demonstrated a best-point specific fuel consumption of 0.39 to 0.41 lb/bhp-hr. Larger Dynastar compression-ignition engines would have a lower specific fuel consumption in the range of 0.40 to 0.42 lb/bhp-hr.

Development work by the Thiokol Chemical Corporation on Dynastar engines has to date been accomplished with corporate funds.

CONCLUSIONS

Projected development programs of the Thiokol Chemical Corporation to improve and increase the potential of the Dynastar engine include the following items:

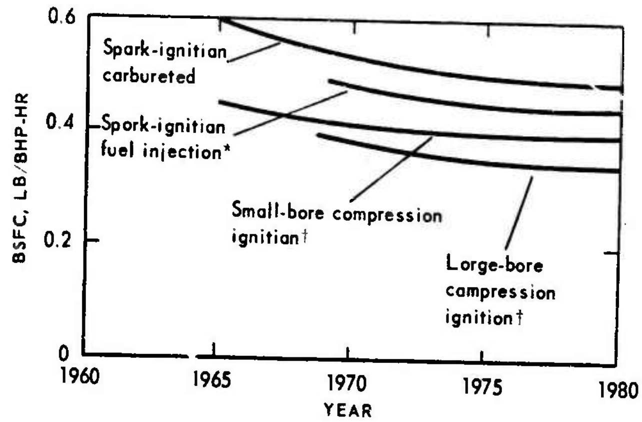


Fig. I-111—Trend Forecast of Fuel Consumption for the Dynastor Engine

*Direct cylinder fuel injection. †Unsupercharged.

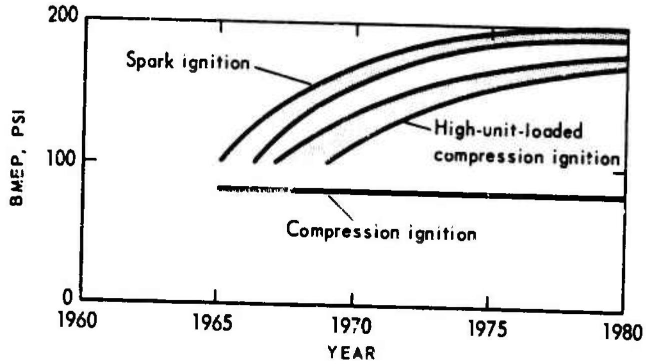


Fig. I-112—Trend Forecast of BMEP for the Dynastor Engine

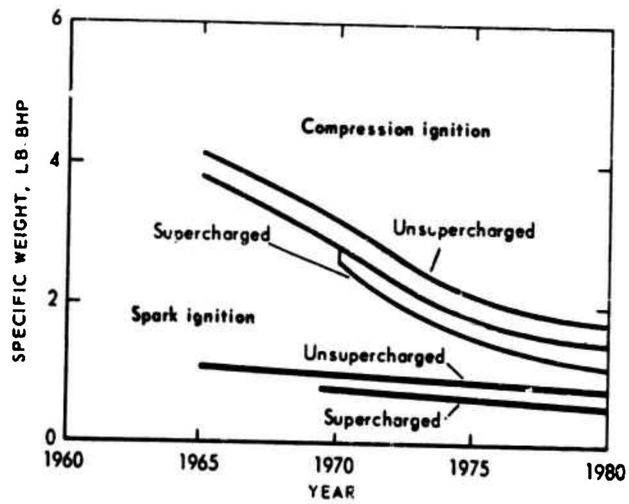


Fig. I-113—Trend Forecast of Specific Weight for the Dynastor Engine

(a) A multifuel capability
(b) An improved combustion system
(c) A hybrid combustion system
(d) Turbosupercharging and turbocompounding
(e) Increased engine speeds
(f) Increased bmep
(g) Air cooling

The primary application of some of the desired improvements (such as items a and f above, and, to a lesser extent, items c, d, and g) would be for military use and would require funding by the military to ensure achievement of the stated objectives within a reasonable time.

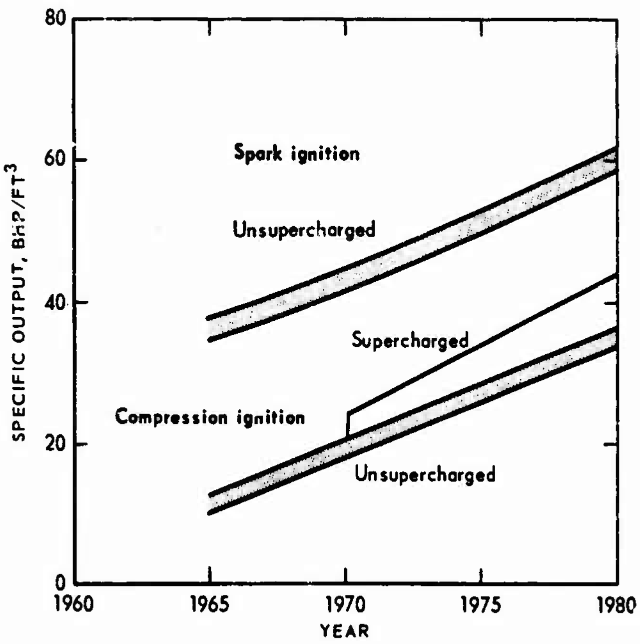


Fig. I-114—Trend Forecast of Specific Output for the Dynastar Engine

The successful development of the Dynastar engine will provide the military with the following engine improvements:
(a) Reduced fuel consumption
(b) Increased power output at reduced weight
(c) Increased power output for engine size (volume)
(d) Multifuel capability

The trend-forecast charts (Figs. I-111 to I-114) illustrate the estimated level of technological achievement with regard to the Dynastar engine through 1980.

Cooperative Government support of Thiokol Chemical Corporation R&D could produce a spark-ignition or compression-ignition Dynastar engine that could improve the physical and performance characteristics of most future tactical vehicles.

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Chapter 8

STIRLING-CYCLE ENGINES

The Stirling-cycle engine is a reciprocating external-combustion engine invented by Robert Stirling in Scotland in 1816. A schematic diagram of the Stirling-cycle engine is shown in Fig. I-115.[1]

The Stirling-cycle engine is unique in that the fuel is ignited in a combustion chamber outside a cylinder to provide the heat to expand gas within the cylinder. The expansion of the gas within the cylinder reciprocates the displacement piston. The engine differs from modern reciprocating engines in that fuel burning is continuous and the products of combustion never enter the cylinder cavity. Heat from the combustion chamber heats the working medium, usually helium or hydrogen within heater tubes, and expands the gas to produce pressure in the cylinder. To continue the cycle the working medium is cooled by water that absorbs the heat dissipated at the radiator. A regenerator stores the heat removed, which would otherwise be lost from the working medium during the constant-volume cooling processes, and returns the heat to the working medium during the constant-volume heating processes. The power piston compresses and expands the working medium at the proper time and the force is transmitted from the power piston to a drive mechanism by means of a power-piston rod.

The major components of the Stirling-cycle engine, the requirements of the working medium, and the specific function or action of the major components are listed in Table I-14. Most hydrocarbon fuels can be used in the Stirling-cycle engine.

Material used for analysis of the Stirling-cycle engine was obtained through a literature search that included publications of the Society of Automotive Engineers (SAE) and various reports and periodicals. Further, personal interviews were conducted with personnel of USAERDL, Ft Belvoir, Va.; ATAC, Warren, Mich.; and the General Motors Research Laboratories, Warren, Mich.

Development of the modern Stirling-cycle engine was pioneered by Philips Laboratories, Eindhoven, Holland. A version of the Stirling-cycle engine was produced by Philips Laboratories in 1937 to serve as a source of power for small electrical generators. This type of engine was chosen because it was relatively free of noise and vibration. The generators then provided power for radios in remote areas.[2]

In 1958 GMC entered into a joint agreement with Philips Laboratories to conduct further research in development of the Stirling-cycle engine.[3]

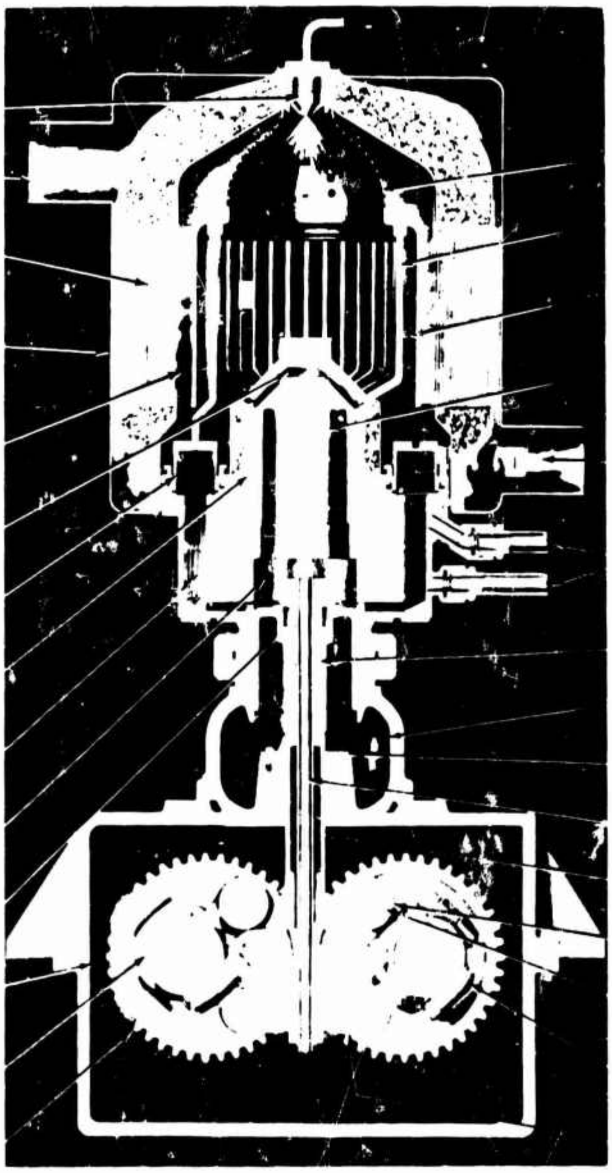

Fig. 1-115—Schematic Diagram of Stirling Thermal Engine[1]

Subsequently various 1-, 2-, and 4-cylinder engines, ranging from 6 to 360 hp, were produced. These experimental models were designed to improve engine fuel efficiency; efforts were not made to reduce engine weight and bulk. The results of tests[1] made on two of the Philips experimental engines are presented in Table 1-15, where the weight and fuel efficiency of the Stirling-cycle engine are compared with that of present comparable spark- and compression-ignition engines.

Although GMC was reluctant to disclose recent findings regarded as proprietary information, certain information was obtained from USAERDL.

TABLE I-14

Operating Principles of the Stirling-Cycle Engine

Major component	Specific function of component
Heater	Heats the working medium; fuel burning in the external combustion system keeps the heater hot; any heat source having a high temperature will cause the heater to operate
Regenerator	Stores heat removed from the working medium (fluid) during the constant-volume cooling process; returns heat to the working medium during the constant-volume heating process
Cooler	Keeps working medium cool by circulating water in annular tubes around it
Displacement piston	Removes and stores energy from regenerator; controls movement of working medium through heater, regenerator, and cooler
Power-piston rod	Effects compression and expansion of the working medium at the proper time

TABLE I-15

GMC Tests on Two Stirling Engines, 1959[4]

Engine category	Fuel type	Fuel heating value, Btu/lb	Maximum economy data (minimum BSFC, lb/bhp-hr)	Full-load data (BSFC, lb/bhp-hr)	Specific weight, lb/hp
Small-Engine Comparison					
Stirling	Diesel no. 1	18,200	0.47	0.528	14.7
Automotive, 4-cyl.	Gasoline	19,160	0.458	0.51	4.68
Large-Engine Comparison					
Stirling	Diesel no. 1	18,200	0.358	0.418	11.0
Diesel, 6-cyl.	Diesel no. 1	18,200	0.40	0.41	10.4
Automotive, V-8	Gasoline	18,900	0.415	0.468	2.80

DISCUSSION

Additional information released by USAERDL personnel disclosed that USAERDL, beginning in 1959, entered into contracts with GMC for the design and construction of three small Stirling-cycle motor-generator sets.[5] The sets were built and tested at the contractor's plant. One set was operated for about 14 hr, when tests were discontinued because of fuel-combustion difficulties. The second set was operated for 40 hr with only minor adjustment, then delivered to USAERDL for endurance tests. This set was operated for an additional 500 hr at USAERDL, where progressive engine deterioration and loss of power and fuel efficiency were evidenced. This engine weighed approximately 154 lb and produced a net 5.2 hp. The complete engine assembly, with accessories, occupied approximately 15 ft^3 of space. Fuel consumption after a break-in period averaged 0.60 lb/bhp-hr.

The third Stirling-cycle motor-generator set was retained by GMC for future corporate development and evaluation. Recently two new Stirling-cycle motor-generator sets were produced by GMC and delivered to USAERDL for test. To date the tests have not been performed, and hence test results are not available.

An important concept—the Dineen process—was introduced by the Fairchild Hiller Corporation in a modified version of the Stirling-cycle engine. This process simplifies the method of heat transfer between the combustion chamber and the working medium. The concept, as discussed by ATAC personnel, will result in a simplified design that should be less costly to produce than the Philips-GMC units that have been built. At present ATAC has a contract with Fairchild Hiller for one Dineen-process 6-hp engine that will be tested and evaluated. This modified engine will be air-cooled, weigh approximately 35 lb, and occupy approximately 1 ft^3 of space. The Fairchild Hiller engine will be capable of operation with various hydrocarbon fuels. The working medium will be air rather than hydrogen or helium.

Interest has been shown in the Stirling-cycle engine as a means of providing a vehicle with the power and fuel efficiency of comparable gasoline or diesel engines at a reduced noise and vibration level.

CONCLUSIONS

The objectives—low noise level and relative freedom from vibration—are considered feasible since fuel in the Stirling engine is burned at a steady rate rather than being expended in a series of explosions, and dynamic balance of the drive mechanism may be readily achieved.

The successful development of the Stirling-cycle engine could produce a power source having a particular use in small vehicles where quiet operation is essential. Therefore it appears that the Government should continue R&D of Stirling engines until the successful development of these models for tactical vehicles has been achieved, unless it is concluded during future engine development that deficiencies encountered cannot be resolved with a reasonable amount of time and effort.

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Chapter 9

BATTERIES

DISCUSSION

Batteries are energy-conversion devices that convert stored energy into electrical energy by a chemical process. They were sometimes used to propel vehicles in the past and are even now used in certain special-purpose vehicles. Some of the advantages of batteries are their silent operation, reasonable cost of operation, good reliability, and relatively easy maintenance.

The basic battery types are nonrechargeable and rechargeable. Nonrechargeable batteries are classified as primary and rechargeable batteries as secondary.

Primary batteries are applicable when low power levels are required, since their weight-to-power ratio is low. These batteries normally have a very short life under high power loads because they cannot be recharged. Therefore they are not considered practical as a power source for tactical vehicles.

Secondary batteries are applicable as a power source for tactical vehicles since their power levels are higher than those of primary batteries and they are rechargeable. Most of today's more promising secondary batteries are shown in Table I-16.[1,2] This table lists both conventional and electrochemical-couple types of batteries. The former is self-contained in a unit package; the latter requires an external source of fuel and accessories to make up a system. The table compares the present capabilities of the batteries with what should be their future potential (in 10 to 15 years) if full effort for their development is continued.

Electrochemical Couples

The most promising of the secondary batteries are the electrochemical couples. The best of these are described here.

Lithium-Nonaqueous-Electrolyte Ambient-Temperature System. If fully developed, this system appears to have the best chance of achieving the highest energy density and highest efficiency of all the electrochemical couples. Power is available immediately and the system self-discharge rate is low.

Air-Zinc System. The air-zinc system, now under active development, uses an air oxidant that is readily available. It does not need to operate at elevated temperatures.

TABLE I-16[1,2]

Types of Secondary Batteries

Battery type	Energy density					Operating life, months	Cycles	Operating temp. range
	Whr/lb			Whr/ft^3				
	Oper.	Future	Theor.[a]	Oper.	Future			
Conventional Batteries								
Lead-acid	15	25	115	1728	2,868	120	400	-40 to 140°F
Nickel-iron	10	12	138	1728	2,073	120	2000	-10 to 115°F
Nickel-cadmium	15	25	107	1728	2,868	96	2000	-40 to 140°F
Silver-zinc	50	100	208	6048	12,096	18	200	0 to 140°F
Silver-lead	17	20	550	4147	4,851	36	500	-10 to 140°F
Electrochemical Couples								
Lithium anode								
Secondary[b]								
Lithium-silver	90	120	230	3801	5,066	18	50-200	Up to 165°F
Secondary[c]	—	—	—	—	—	—	—	—
Lithium-copper	100	200	746	—	9,434	18	50–200	Up to 165°F
Air-zinc	55	100	435	5184	9,434	18	200–400	0 to 165°F
Air-cadmium	30	60	290	—	—	24	—	0 to 160°F
Lithium-chlorine (molten electrolyte)	215	225	700	—	—	24	50–100	650°C

[a]Active materials only.

[b]Lithium-silver chloride with lithium chloride–aluminum chloride salts dissolved in propylene carbonate system.

[c]Lithium-copper fluoride with sodium hexafluorophosphate in propylene carbonate system.

TABLE I-17

Comparison of Batteries and Spark-Ignition Engine, Including Fuel and Accessories

Type	Estimated weight,[a] lb		Estimated volume,[a] ft^3	
	Present	Future	Present	Future
Lead-acid	40,000	24,000	347	209
Silver-zinc	12,000	6,000	97	49.6
Air-zinc	10,909	6,000	115	42.5
Lithium-chlorine[b]	8,340	5,006	nd[c]	nd[c]
Military spark-ignition engine[d]	5,160	4,644	97	87

[a]Weight and volume based on 10 hr of operation at 600 kw.

[b]Based on 100 whr/lb; future at 225 whr/lb.

[c]Not determined.

[d]Engine, including fuel and cooling for 600 hp, estimated at 3.6 lb/hp, 15 hp/ft^3, and 0.5 lb/hp-hr BSFC.

Air-Cadmium System. This is another system currently under development. Like the air-zinc system, it uses an air oxidant that is readily available and does not need to operate at elevated temperatures. In addition it has a good recharging rate.

Molten-Electrolyte System. The molten-electrolyte system, also being developed now, has the potential to achieve a very high energy density. It also has an excellent recharging rate.

Comparison with Spark-Ignition Engine

A weight and volume comparison is made in Table I-17 between the most promising conventional and electrochemical-couple batteries and the military spark-ignition engine. Anticipated improvements are given also. On the basis of this comparison, batteries do not appear to offer any advantage over spark-ignition engines for the present or the near future.[3]

CONCLUSION

Batteries are unable to operate at high current levels for extended periods, have low energy output for their weight and volume, and are expensive for the limited number of operating cycles. They offer only one means of improving the operating capability of tactical vehicles: their silent operation. Therefore R&D effort should be expended by the Government on batteries as a source of power for tactical vehicles only if silent operation is a prime requisite.

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Chapter 10

FUEL CELLS

DISCUSSION

The principal feature of a fuel cell is its ability to convert the potential energy in liquid or gaseous fuels to electrical energy through a chemical reaction.

The primary reason for interest in the extensive R&D of fuel cells is that theoretically 90 percent of available heat energy can be obtained by converting the energy in fuel to electrical energy, as compared to 60 percent from fuels employed in an internal-combustion engine.

If this theoretical efficiency could be attained, or even nearly approached, the use of hydrocarbon-fueled fuel cells would notably reduce the logistic problems now associated with hydrocarbon-fueled internal-combustion engines.

Common forms of fuel are hydrogen, methanol, ammonia, and hydrocarbons.

On the basis of weight, hydrocarbons have more than twice the obtainable energy than any other fuel actively considered except hydrogen, which is $2\frac{1}{2}$ times better than hydrocarbon fuels. This advantage is lost since hydrogen in this application must be stored in heavy pressure containers.

On the basis of volume, hydrogen has a lower energy content than any of the other fuels. Therefore, on the basis of fuel efficiencies theoretically obtainable, development of fuel cells using hydrocarbons to replace internal-combustion engines appears encouraging.

Hydrocarbon fuels produce electrical power by chemically combining the hydrogen in hydrocarbons with oxygen. Although fuel cells' efficiencies and power densities are higher with oxygen than with air, it is assumed that air is the only practical source of oxidant for fuel cells. When air is used as an oxidant large amounts of nitrogen are produced that must be rejected from the cell.

Development effort through 1965 had been limited to laboratory models, vehicle prototypes, and spacecraft application. Laboratory models have used exotic fuels and materials without limitations of cost, space, or weight, in order to produce the best efficiencies. The best hydrocarbon-air fuel-cell efficiencies under laboratory conditions are at present only equivalent to the efficiency of an average compression-ignition reciprocating engine. However, fuel cells employing ammonia or hydrogen obtain considerably better efficiencies than average compression-ignition reciprocating engines (see Fig. I-116).

A prototype hydrocarbon-air fuel cell installed in a tractor proved larger and heavier than a reciprocating engine for the same vehicle would have been. In addition, fuel efficiencies achieved by the fuel cell were lower than could have been achieved by a reciprocating engine. USAERDL at Ft Belvoir are currently incorporating a fuel cell into a ¾-ton-truck test bed. Indications are that this fuel cell will occupy more space than the reciprocating engine it will replace and will increase the vehicle's weight.

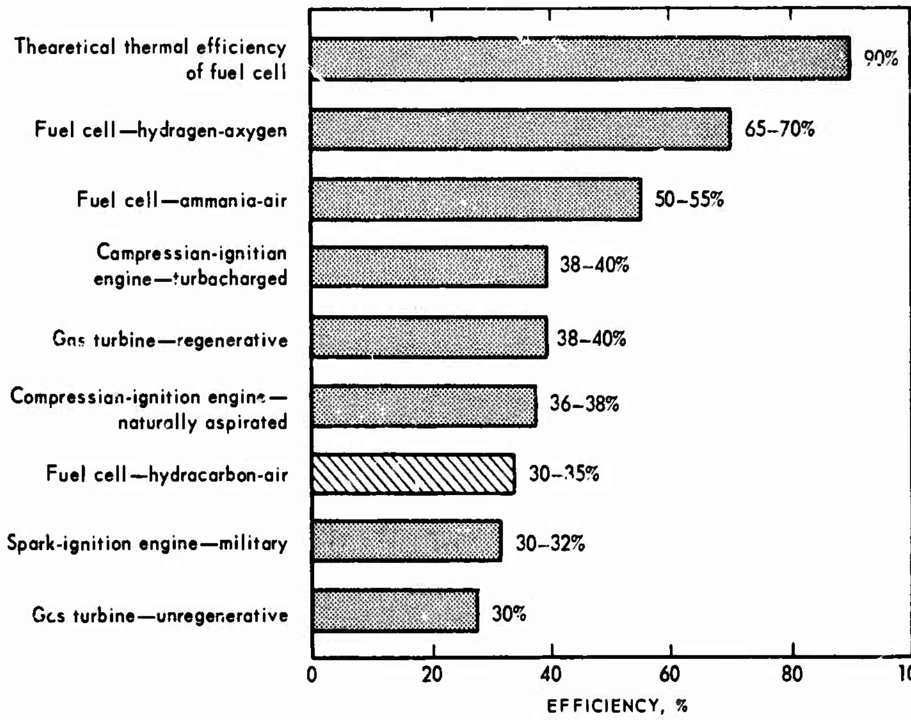

Fig. I-116—Efficiency Comparison of Selected Engines

Fuel-cell installations in spacecraft have been acceptable since cost has not been a primary consideration. The fuel has consisted of purified hydrogen and oxygen that produced water, a by-product essential to the crew in flight. Platinum has been the only catalyst employed to produce the required chemical reaction.

Many other types of fuel cells have been installed in special-purpose vehicles such as forklift trucks for the purpose of obtaining research data that could be used by the development engineer. Oxygen would produce a more efficient and more compact fuel cell than air, but at present no compact and efficient method has been devised to separate oxygen from air. To separate oxygen from air it is found that the cost is exceedingly high, the oxygen must be contained in pressurized tanks that are heavy and bulky, and the oxygen-separating system is relatively complex.

Present hydrocarbon fuels contain impurities such as sulfur compounds that greatly reduce the useful life of most catalysts. This problem is reduced if a noble metal such as platinum is used. However, the world supply of platinum is limited, and its cost is prohibitive. Even if cost were not a consideration, the available supply of platinum would not equal the amount required to replace present reciprocating engines with fuel cells. Thus the problem of finding an efficient, less expensive, and more abundant catalyst remains to be solved.

The development of fuel cells for application in tactical vehicles could provide the following advantages over presently available engines.

(a) They need not be localized and therefore could be installed at optimum locations.

(b) They require very little maintenance since there are no moving parts.

(c) They operate silently, thus lessening the likelihood of enemy detection.

(d) They generate little heat, thereby lessening enemy infrared detection.

(e) They can withstand 100-percent overload for brief periods of time.

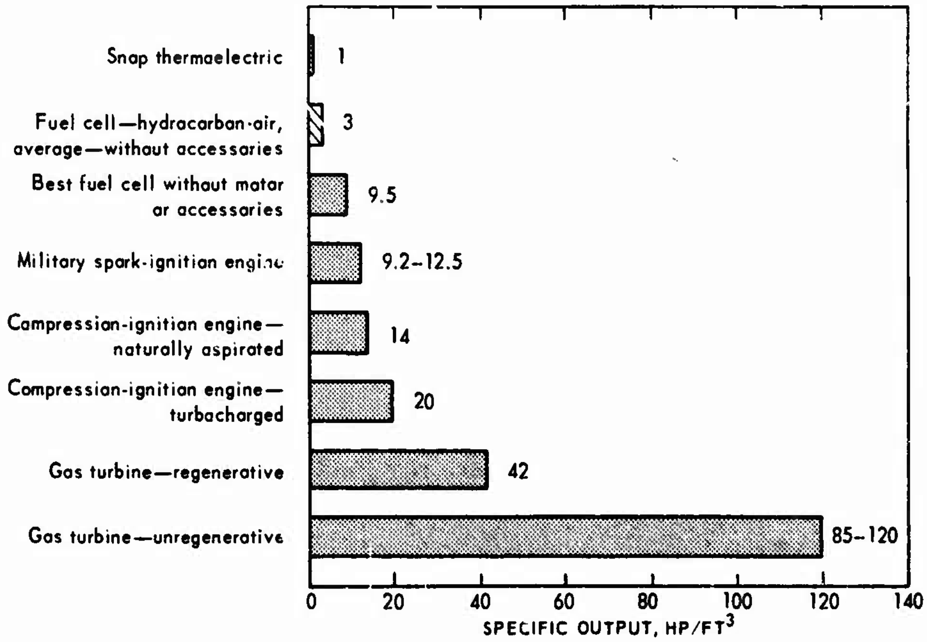

Fig. I-117—Specific Output Comparison of Selected Engines

Analysis of all available data indicates that the volume (see Fig. I-117) and weight (see Fig. I-118) of a hydrocarbon fuel cell are considerably greater than those of present comparable spark-ignition and compression-ignition reciprocating engines. No feasible approach or concept has been proposed that would overcome this disadvantage. Also, as indicated previously, readily available fuels do not produce the required power in fuel cells at efficiencies greater than those of currently available internal-combustion reciprocating engines.

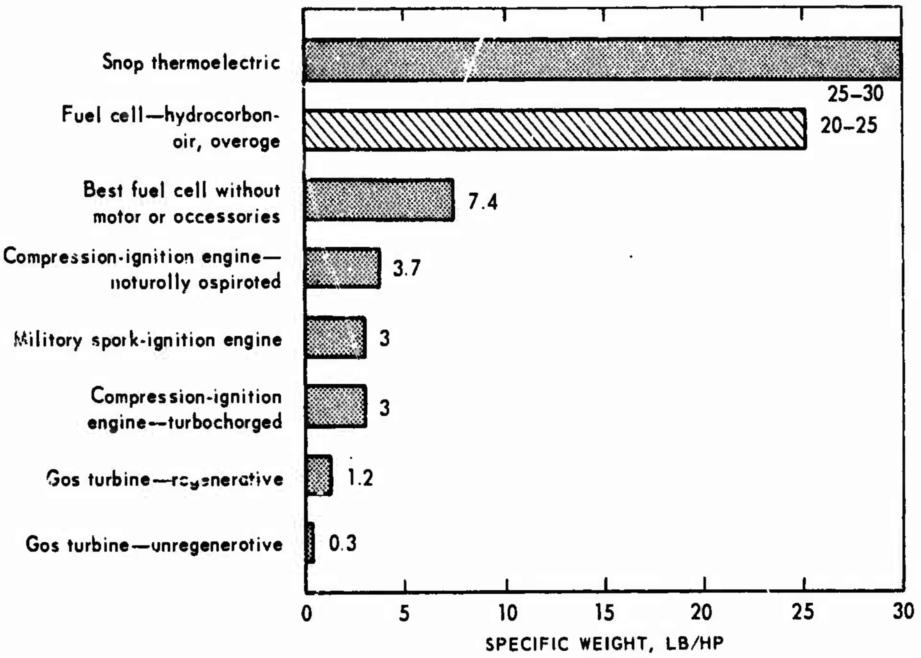

Fig. I-118—Specific Weight Comparison of Selected Engines

CONCLUSIONS

It appears feasible that hydrocarbon fuel cells will find acceptance for application in stationary equipment where increased weight and size are not prime factors of importance. However, fuel cells will continue to be too heavy, bulky, and inefficient within the foreseeable future for use in tactical vehicles. Therefore R&D should be held in abeyance until acceptable fuel cells are developed for stationary equipment.

If fuel cells in the future do become practical for stationary equipment and if many of the present problems are resolved, R&D for tactical-vehicle applications should then be reconsidered.

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Chapter 11

UNIQUE ENERGY-CONVERSION DEVICES

Unique energy-conversion devices were evaluated with respect to their suitability for propulsion of tactical vehicles. These energy-conversion devices can convert either thermal, chemical, or mechanical energy directly into electrical energy. The advantages they may offer are:

(a) Relatively silent operation
(b) Ease of operation
(c) Simple construction and few moving parts
(d) Adaptability to use potential energy from waste heat as in total-energy systems

The energy-conversion devices that were considered are:

(a) Thermoelectric generators
(b) Thermionic generators
(c) Electrostatic generators
(d) Photovoltaic generators
(e) Thermophotovoltaic generators
(f) Magnetostriction generators
(g) Pulse-power sources
(h) Hall-effect devices
(i) Magnetohydrodynamic generators

DISCUSSION

Substantial R&D effort has been and still is being made by both industry and government to develop unique energy-conversion devices, mainly for space applications. In 1947 a complete survey of all known power sources was made by Armour Research Corporation.[1] At that time it was determined that all unique energy-conversion devices had very low operating efficiencies. The power-output levels were found to be too low for their weights and sizes for practical application in vehicles.

An effort was made to update these findings by contacting various industries and Government agencies. The updated findings follow.

Thermoelectric Generators

The thermoelectric generator operates on the principle that a current will flow in a circuit consisting of two different materials and two junctions maintained at different temperatures. Thermal energy is converted directly into electrical energy.

Many generators have been built incorporating this principle and today are finding widespread use in control and safety applications. In the past, thermoelectric generators were characterized by their large weight and volume per watt of output. Efficiency ranged from a low of 0.2 to 8.2 percent and a 1-kw machine would weigh about 1 ton.[1] With the development of semiconductors, these devices began to show more promise, and efficiencies of 3 to 7 percent are easily attained. Typical power-density values for a 10-kw system burning liquid fuel are 0.0133 kw/lb and 0.503 kw/ft^3 and an energy-conversion efficiency of 7.13 percent.

Laboratory models have attained an efficiency of 13 percent and weight and volume have been reduced. The problem areas are heat shunting, thermal stress, high electrical resistivity, and low efficiency, but improvements are expected in the future. However, this means of producing electrical energy is not expected to be used for propulsion because its cost is high, its efficiency is low, and its weight and volume are much too high for its output.

Thermionic Generators

Thermionic generators are basically emission devices that convert heat energy into electrical energy. One plate or cathode is heated, forcing electrons to be emitted. A cooler collector or plate collects the electrons. If this plate is connected to an external electrical circuit, electrical power is obtained.

Power outputs of 5 w/cm^2 with an efficiency of 10 percent are common. Higher power outputs have been demonstrated in the laboratory, and the overall efficiency can be expected to reach 15 to 18 percent.

These devices have poor efficiency, low power levels, and high weight and volume. They will find increased application in stationary equipment in conjunction with nuclear reactors.

Electrostatic Generators

Electrostatic generators take advantage of phenomena that are due to the attraction or repulsion of electric charges but are not dependent on their motion. These devices are capable of delivering a few kilowatts at a few megavolts. Since they are primarily high-voltage—low-current devices, applications for vehicle propulsion are not practical.

Photovoltaic Generators

The photovoltaic and thermophotovoltaic generators both operate on the same principle: if one of two similar electrodes immersed in a suitable electrolyte or other substance is treated with a sensitive material and illuminated by a light source, a voltage is generated. If an external load circuit is added, a small current will flow.

The photovoltaic generator is normally associated with the conversion of solar energy whose spectral intensity peaks at about 0.55μ. The theoretical area requirement is 10 ft^2/kw, but, for a 10 percent conversion efficiency, 100 ft^2/kw is required. The power density is approximately 10 w/ft^3 and 2 w/lb.

Solar energy is a free inexhaustible source, but it is not available at all times, and for this reason photovoltaic generators cannot be used for propulsion of land vehicles. The high volume requirements, low power levels, and low efficiencies are other major reasons why photovoltaic generators are not practical for vehicle application.

Thermophotovoltaic Generators

The thermophotovoltaic generator converts energy from near-infrared radiant energy at a wavelength of approximately 2.5μ. A practical converter burns liquid fuel in an inner cylinder, thus producing radiant energy that strikes the surrounding cooled, thin-film photovoltaic cell. A reflector returns unabsorbed radiant energy to the heat source.

A power output of 0.4 w/cm^2 of cell area is now being achieved, and it may be possible to achieve 1.0 w/cm^2.

Research is now being conducted on a 500-watt backpack unit in which efficiencies of 3 percent for air-cooled units and 5 percent for water-cooled units have been achieved and efficiencies of 8 percent and 15 percent respectively are anticipated. The weight is estimated[2] to be 40 lb with a volume of $1\frac{1}{2}$ ft^3. The power density will be 12.5 w/lb and 333 w/ft^3.

The low efficiencies, low power levels, and high weight and volume requirements of thermophotovoltaic generators do not lend themselves to propulsion of vehicles.

Magnetostriction Generators

Magnetostriction generators take advantage of the phenomenon that occurs when certain magnetized materials are either stretched or compressed: the materials modify their magnetic fields in such a way that a coil placed in the magnetic field will produce electrical energy. Since the amount of power that can be generated by magnetostriction is very low, no further consideration need be given this device for application in vehicles.

Pulse-Power Sources

Pulse power by definition means a series of timed pulses of energy usually converted from another form of energy. For example, potential energy can be activated to controlled explosive forces as a means of propulsion. Pulse duration can vary from a submicrosecond to many milliseconds.

Short bursts of power from controlled nuclear explosions are being investigated for space missions. Although these devices have found applications in missiles, they are not suitable for use in tactical vehicles.

Hall-Effect Devices

The Hall effect is a difference of potential observed between the edges of a strip of metal carrying current in a longitudinal direction when placed in a magnetic field perpendicular to the plane of the strip. It occurs in bismuth, germanium, and to a lesser extent in other metals.

Devices relying on this effect have been used in conjunction with other devices to obtain small amounts of power, but they are not applicable to propulsion of vehicles.

Magnetohydrodynamic Generators

The magnetohydrodynamic (MHD) generator, also known as a magneto-plasmadynamic generator, produces electricity by moving a hot ionized gas stream through a magnetic field, cutting the lines of force. Electrodes located in the gas duct pick up the generated power and conduct electricity to an external load circuit.

One of the basic problems of the MHD generator is that high gas temperatures and good conductivity are required to obtain ionization. Several methods are available to increase conductivity and reduce the operating temperatures. One method to increase conductivity is the use of a strong magnetic field to induce nonequilibrium ionization. Other methods to increase conductivity are preionization by an electron beam or photoionization. In the former method, two electron-beam guns located on each side of the duct shoot electrons into the partly ionized gas to increase ionization. In the latter, light from an ultraviolet lamp is shot into a generator duct through quartz ports to augment ionization.[3]

The results of a study[4] indicate that the efficiency of a fossil-fueled power plant may be raised from 40 to 55 percent by passing the hot furnace gases through an MHD generator prior to entering the boilers.

An early prototype MHD generator[5] has produced 2.8 Mw. The design goal is 35 Mw for 3 min.

Experiments with new superconducting magnets have resulted in a weight reduction in a specific application from 5 tons to 600 lb[6]. The Government is conducting active R&D on these devices, and they are intended for operation in conjunction with large nuclear reactors. A 350-kw SNAP 50 reactor is now under development. Another 20 years of research is expected before the MHD nuclear reactor will become operational[3]. Because a nuclear reactor is required to provide a continuous source of energy, these generators are not suitable for propulsion of tactical vehicles.

SUMMARY

Table I-18 presents a compilation of data on the various unique energy-conversion devices, including their power sources, power densities, and power levels. The values in the table represent figures based on optimum conditions. An examination of the table reveals that the thermophotovoltaic generator has the highest power density in watts per pound.

A weight and volume comparison of this device with a conventional military spark-ignition engine is shown in Fig. I-119. The thermophotovoltaic generator produces considerably less horsepower per cubic foot than the military spark-ignition engine, and its weight per horsepower is considerably higher. Projected increases in efficiency will not appreciably increase the specific output or reduce the specific weight of this device to a degree where it can compete with the present military spark-ignition engine.

TABLE I-18

Unique Energy-Conversion Devices

Device	Power source	Power density		Efficiency, %		Power level
		Watts/lb	Watts/ft^3	Field conditions	Laboratory conditions	
Thermoelectric generator	Liquid fuel	13.3	503	7.13	15	10 kw
	Solid fuel	0.346	5.79	0.4		8 w
	Sun			4		1 kw
	Nuclear fission	10[a]	207[a]	4–4.5		1 kw–3 mw[n]
Thermionic generator	Liquid fuel	2–3	4645	10	15–18	
Electrostatic generator	Mechanical devices	12.5	333	5	15	2–3 kw at 2–3 megavolts
Photovoltaic generator	Light	2	10	10		
Thermophotovoltaic generator	Liquid fuel, Nuclear fission	125	333	3–5	8–15	
Magnetostriction generator	Mechanical devices	na[b]	na	na	na	Very low
Pulse-power source	Chemicals	na	na	na	na	Intermittent
	Nuclear fission	na	na	na	na	Intermittent
Hall-effect device	Magnetic field	na	na	na	na	Very low
Magnetohydrodynamic generator	Argon gas	nd[c]	nd	nd	nd	11.2 mw for a few seconds
	Nuclear fission	nd	nd	nd	nd	

[a]Estimated.
[b]Not available.
[c]Not determined.

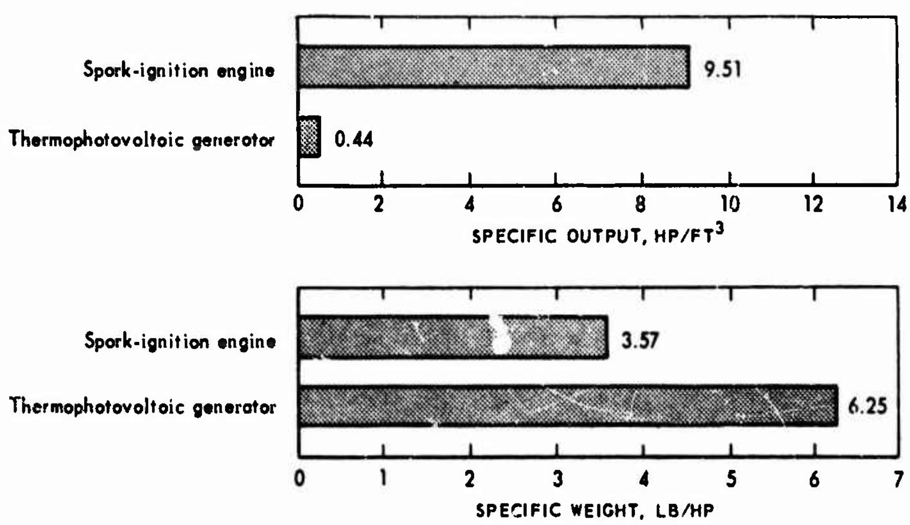

Fig. I-119—Comparison of Thermophotovoltaic Generator with Military Spark-Ignition Engine

CONCLUSIONS

All unique energy-conversion devices considered are much too heavy, require too much space, and are too inefficient for application in tactical vehicles. Development within the foreseeable future would not result in an energy-conversion device that would improve the capabilities of tactical vehicles.

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Chapter 12

NUCLEAR PROPULSION

DISCUSSION

Nuclear reactors were considered for application in tactical vehicles since they could greatly reduce the requirements for refueling. If nuclear reactors in vehicles were practical, they would substantially reduce petroleum, oil, and lubricant logistics.

Heat, the primary product of a nuclear reactor, can be converted to mechanical energy by two methods: One method is to heat a medium such as water to operate a steam engine; the other is to convert heat to electrical energy by means of a suitable energy-conversion device.

A study completed in 1960 by the Operations Research Office of Johns Hopkins University (ORO), now RAC, evaluated the possibilities of nuclear reactors for use in combat vehicles.[1]

A complementary study was completed in 1961 by ORO to determine the operational feasibility of powering combat vehicles with nuclear reactors.[2]

A study completed in 1962 by RAC analyzed the possible application of nuclear reactors to vehicles by the use of direct-conversion devices.[3]

Evaluation of our findings was based on the cited reports, which are classified Secret. To preclude such a classification of this report, an oral presentation of pertinent details was made to the Project Advisory Group, and consequently they are not included in this document. The conclusions presented below are unclassified.

CONCLUSIONS

Although it may be technically feasible to develop nuclear reactors for application to tactical vehicles, their extremely high cost and large size and weight for their output, combined with the possibility of personnel being exposed to radiation hazards, would preclude their use as power sources. Therefore R&D effort for nuclear reactors in tactical vehicles is not warranted until more efficient shielding materials become available and the cost of the reactors is substantially reduced.

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Chapter 13

GAS-TURBINE ENGINES

INTRODUCTION

The basic concept of the gas-turbine engine dates back many years, but it was just before WWII that the first practical aircraft gas turbines were developed in Europe. The US military establishment realized the future potential of the gas-turbine engine and began energetic development programs to produce gas-turbine engines for aircraft propulsion. Because of their success in aircraft propulsion, gas turbines have found application in other fields such as industrial plants, electrical generating plants, total-energy systems, ship propulsion, locomotive propulsion, and just about every other application requiring shaft power with the exception of automotive-vehicle propulsion.

Much interest has been shown in the past 10 years by both industry and the military in the gas-turbine engine as a power source for automotive vehicles. The automotive industry has developed several generations of experimental gas-turbine engines for truck and automobile applications. The military has installed some of these engines in tactical vehicles on an experimental basis. Some earlier experimental vehicle applications of the gas-turbine engine by the military utilized modified aircraft engines. The modified aircraft engines did not demonstrate satisfactory operation in surface vehicles. These installations, which are listed in Table I-19, did, however, provide engine and vehicle designers with valuable information that could be used in designing a gas-turbine engine specifically for the propulsion of surface vehicles.

In 1960 the military contracted with industry for the development of gas-turbine engines for possible use in tank and marine application. The first program was a joint Army-Navy program for the development of a 600-hp turbine engine. The scheduled portion of the program was completed, and Orenda Engines, Ltd., was selected to produce several prototypes. More recently (1966) ATAC contracted with the Lycoming Division of AVCO Corporation to design, develop, and fabricate an advanced-technology 1500-hp gas-turbine engine for application in future main battle tanks. The program is proceeding at time of writing, and major design problems have not been encountered.

TABLE I-19

Experimental Gas-Turbine Power Plants in Military Vehicles

Power plant	Vehicle	Horsepower
USA		
GMC GT-305	LARC V	225
Pratt & Whitney PT-6	LARC V	350
Williams WR-8	M151	72
GMC GT-305	M56	225
GMC GT-305	5-ton truck	225
GMC GT-305	M113	225
Boeing 502-9A	T55	175
Solar 1000MV	T95 MBT	600 (downrated)
Solar 1000MC (3)	Land train (turboelectric)	1050 (each)
Solar 1000MV (2)	LCA	1140 (each)
Lycoming TF 1460	LVHX1	1200
Lycoming TF 20	LVW	1600
Solar 1000MV	LVHX2	1040
GMC GT-305	DW-15 tractor	225
Lycoming T53	Flying DUKW	700
Boeing 502-2E (2)	T42	300 (each)
GMC GT-305	M41	225
Orenda OT4	M48	600
Boeing 502-2E	T42	300
+ 6V53		+200
Foreign		
Boeing 502MA	Swedish S-tank	330
+ Rolls Royce K60	(production)	+240

PRINCIPLES OF OPERATIONS

The gas-turbine engine, essentially a hot-gas-producing machine that receives energy from expanded gases, consists of several major components. Collectively the components perform a cycle of events similar to those that occur in the cylinder of the conventional reciprocating-piston engine. Gas-turbine engines operate on the Brayton or Joule cycle. Atmospheric air is compressed adiabatically, the combustion process adds heat at a constant gas pressure, and the gases are expanded adiabatically through a turbine wheel to provide shaft power. The gas-turbine engine differs from the standard jet-cycle turbine engine, which ejects hot gases through a tailpipe to produce thrust. The jet-cycle turbine engine is shown in Fig. I-120.

Open-Cycle Turbines

The basic simple open-cycle gas turbine, shown schematically in Fig. I-121, is essentially a single-shaft gasifier or gas producer. The gas producer consists of three major components: a compressor, a burner, and a gas producer (compressor drive turbine). The compressor and gas-producer turbine are fixed to a common shaft and are considered to be a single rotating assembly. The simple two-shaft open-cycle gas turbine, also referred to as the "split-shaft" (or "free-spool") turbine, is shown schematically in Fig. I-122.

It consists of two major sections: the gas-producer section, which is identical to that in the single-shaft turbine, and the power section. The gas-producer section consists of three major components, i.e., a compressor, a burner, and compressor turbine. The compressor and gas-producer turbine are fixed to the same shaft and rotate as a single assembly. The power section contains a power-turbine wheel assembly and output reduction gearing.

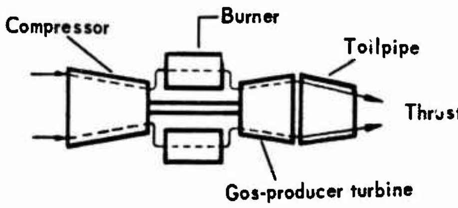


Fig. I-120—Schematic Diagram of Standard Jet-Cycle Turbine Engine

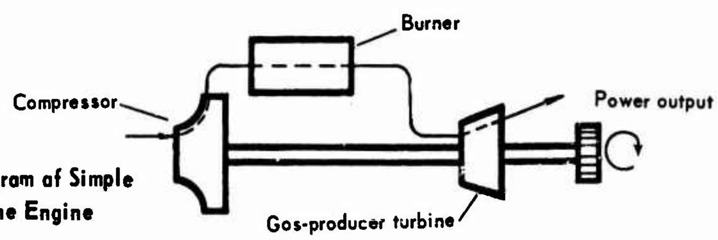


Fig. I-121—Schematic Diagram of Simple Open-Cycle Gas-Turbine Engine

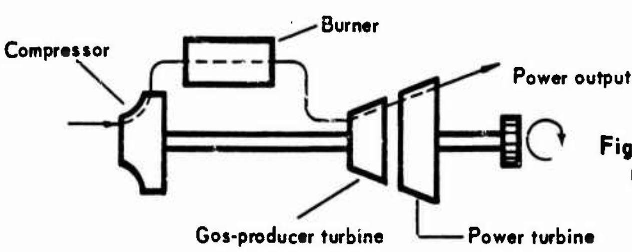


Fig. I-122—Schematic Diagram of Open-Cycle Gas-Turbine Engine with Split Turbine

In a typical simple open-cycle split-shaft gas-turbine engine (shown diagramatically in Fig. I-123 and pictorially in Fig. I-124), the cycle of events in the gasifier section is similar to that of a 4-stroke-cycle reciprocating engine. Air is drawn into the unit by a single- or multistage compressor rotating at speeds up to 50,000 rpm. Engine pressure and temperature are increased because of the work performed. The compressed air is directed through ducting into the combustion chamber, where fuel is injected in the form of a spray, ignited, and expanded in a continuous combustion process. The temperature of the burned gases ranges from 1600 to 2100°F. The hot gases pass from the combustion chamber through nozzles to and through the gasifier turbine wheel driving the compressor. Approximately half to two-thirds of the total gas energy is used to drive the compressor. The still expanding gases are then directed to the power-turbine wheel, rotating it and its reduction-gear set and

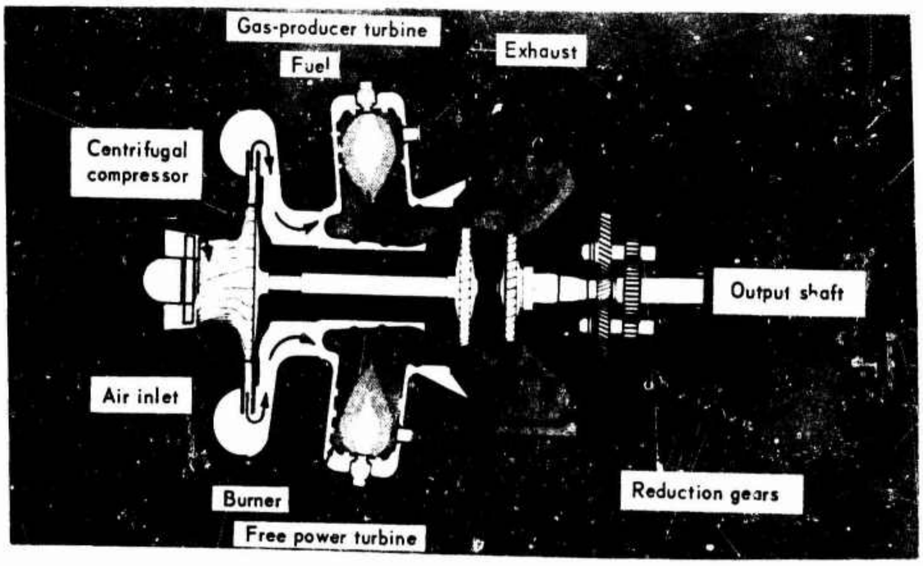


Fig. I-123—Operational Diagram of Simple Open-Cycle Split-Shaft Gas-Turbine Engine

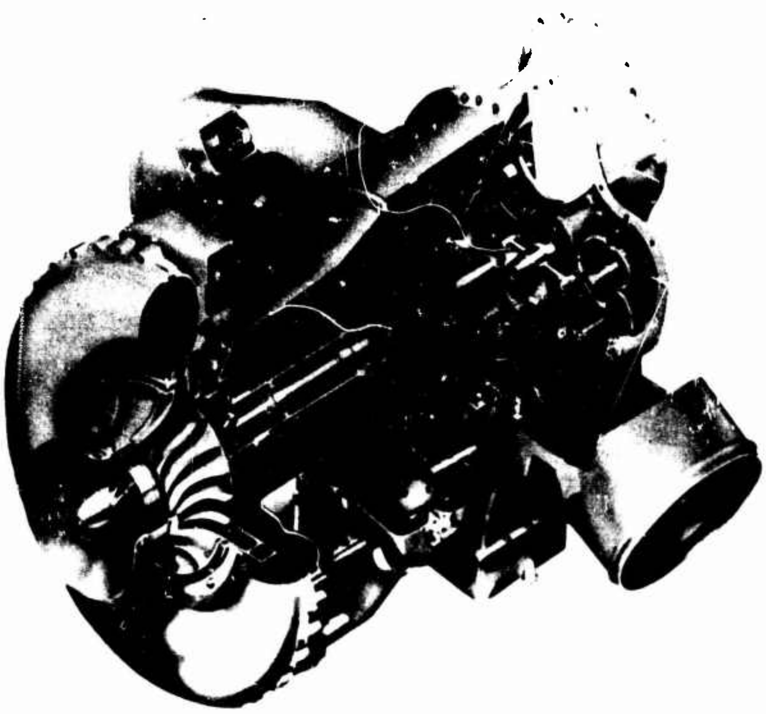


Fig. I-124—Cross Section of Typical Simple Open-Cycle Split-Shaft Gas-Turbine Engine (Boeing) output shaft. The gases are then exhausted to the atmosphere through ducting. The cycle of operation of each component stage in a gas turbine is continuous, not intermittent as in the cylinder of a reciprocating engine. Nonfluctuating continuous rotary motion results.

Because the power-turbine wheel and shaft are not connected mechanically to the gasifier components, the assembly performs in a manner similar to that of a torque converter. As speed decreases, torque increases, and at stall or near stall speed the output torque is equal to approximately three times the torque at maximum speed. The output characteristics of split-shaft gas-turbine engines are ideally suited to the power requirements of automotive-type vehicles. Figure I-125 illustrates a comparison of the torque characteristics of turbine and piston engines. The curves indicate that the turbine engine delivers an increased torque with decreasing engine speed (down to turbine stall speed).

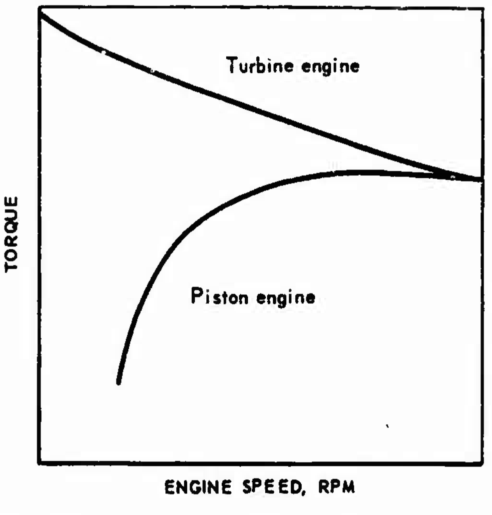


Fig. I-125—Comparison of Torque Characteristics of Turbine Engine and Piston Engine

This is in contrast to the piston engine, which delivers maximum torque at 40 to 60 percent of its maximum speed and power. The ability to provide high torque at stall and near-stall speeds with a completely free power-turbine wheel eliminates the need for a clutch or torque converter. This factor reduces the size and weight of the transmission unit when consideration is given to the complete vehicle power train.

The single-shaft turbine operates in the same manner as the split-shaft turbine, except that the gases are exhausted after passing through the gasifier-turbine wheel. The single-shaft turbine does not have a power-turbine wheel that can rotate at speeds independent of the gas-producer section and hence must be coupled to an infinitely variable transmission system, such as an electric-drive or hydrostatic system.

The fuel consumption of the simple open-cycle gas turbine is much higher than that of the spark-ignition gasoline engine or the compression-ignition engine. The specific fuel rate of the unregenerated gas-turbine engine in the middle speed-power ranges is approximately 50 percent higher than that of the conventional spark-ignition engine and approximately 100 percent higher than that of the compression-ignition engine.

Regenerative-Cycle Turbines

The most effective means of reducing the fuel consumption of the turbine engine is to incorporate a heat exchanger. The heat exchanger recovers some of the exhaust heat by raising the temperature of the air between the compressor and burner sections, as shown schematically in Fig. I-126. Turbines incorporating heat exchangers are referred to as "regenerative-cycle turbines." The heat exchanger may be either stationary or rotating. Stationary heat exchangers are called "recuperators," and the rotary types are called "regenerators."

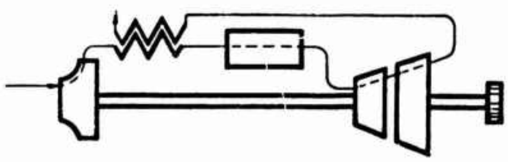


Fig. I-126—Schematic Diagram of Regenerative-Cycle Gas-Turbine Engine

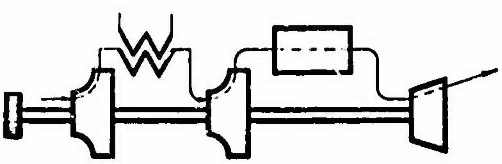


Fig. I-127—Schematic Diagram of Regenerative-Cycle Engine with Two-Stage Compression and Intercooling

To reduce fuel consumption in the part-load power ranges, turbine engines have been designed using various other cycle concepts. Schematic diagrams of these concepts are shown in Figs. I-127 to I-129. These concepts incorporate intercooling or reheat systems, or a combination of the two. However, these cycle concepts tend to become overly complex and costly. Figure I-130 illustrates a comparison of the fuel-consumption characteristics of a typical nonregenerative and regenerative turbine engine. A reduction in fuel consumption of approximately 35 percent is achieved by regeneration. Figure I-131 illustrates fuel-consumption characteristics as a function of percentage of power output for varying recuperator effectiveness in a typical turbine engine. Heat exchangers with high effectiveness have a significant effect on fuel consumption. However, the more efficient heat exchangers are bulkier, heavier, and more costly.

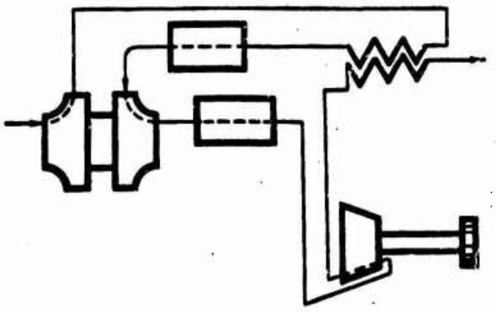


Fig. I-128—Schematic Diagram of Regenerative-Cycle Engine with Reheat Combustor

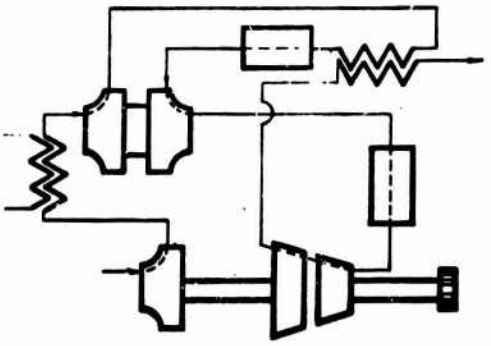


Fig. I-129—Schematic Diagram of Regenerative-Cycle Engine with Reheat Combustor and Two-Stage Intercooled Compression (Turbocharged Cycle)

HEAT EXCHANGERS

There are many different heat-exchanger designs and concepts. The GMT-305 gas-turbine engine, shown in Fig. I-132, incorporates two vertically mounted rotating-drum heat exchangers. The heat exchangers were constructed of a very thin matrix of alternating sheet and corrugated metal. The heat exchanger rotates at approximately 30 rpm.

Figure I-133 illustrates the Chrysler Corporation's Model A-831 experimental automotive regenerative-turbine engine. This unit incorporates two vertically mounted rotary disk-type heat exchangers utilizing a metal matrix

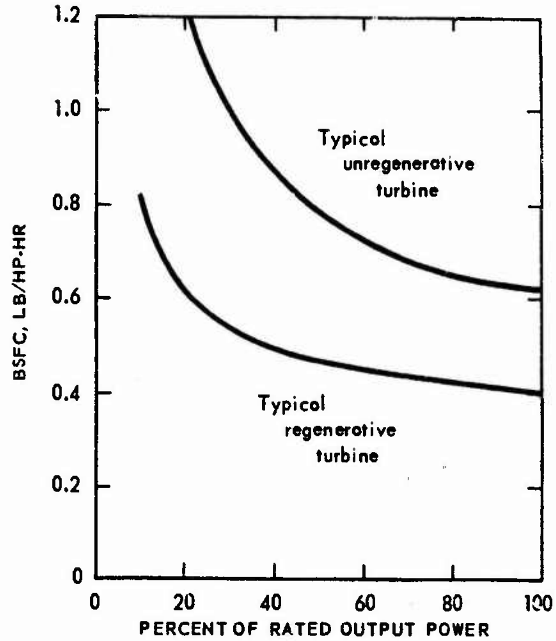


Fig. I-130—Comparison of Fuel Consumption of Nonregenerative and Regenerative Turbine Engines

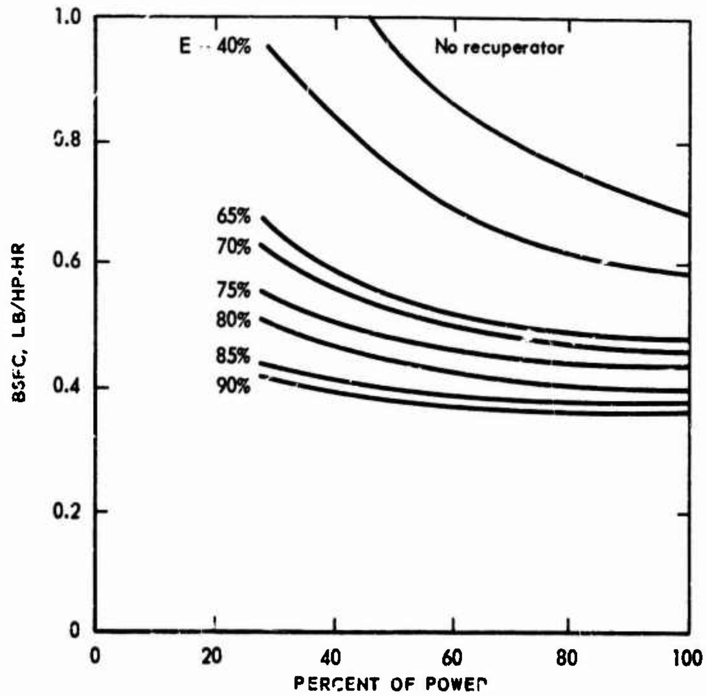


Fig. I-131—Fuel Consumption as a Function of Percentage of Power with Varying Recuperator Effectiveness for a Typical Gas Turbine

Based on nominal cycle temperature of approximately 1800° F.

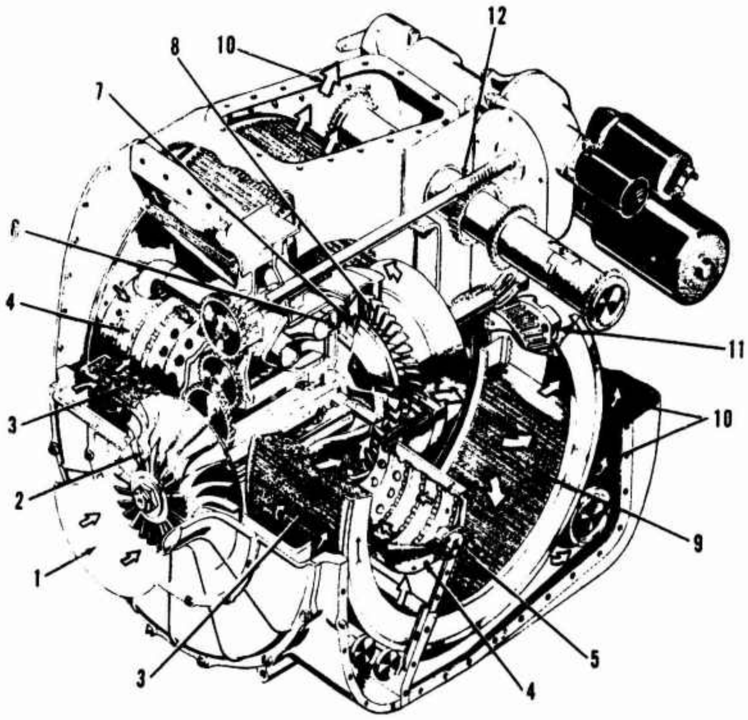


Fig. I-132—Cutaway View of Early GMT-305 Turbine Engine, Showing Twin Vertical Rotating-Drum Metal-Matrix Heat Exchanger

1. Atmospheric air enters the air inlet.
2. The air is compressed by the compressor to above 3 atmospheres of pressure.
3. Rotating regenerators heat the compressed air as it passes through.
4. The heated compressed air enters the combustors.
5. Nozzles inject fuel into the combustors.
6. Gases resulting from the combustion of fuel and air pass through turbine vanes.
7. These gases first drive the gasifier turbine, which powers the air compressor (item 2).
8. The gases then drive the power turbine. (Note that the gasifier turbine and the power turbine are not connected mechanically.)
9. The hot gas exhausted from the power turbine is cooled by the rotating regenerators (item 3).
10. Exhaust gas at 300 to 500°F is directed out of the exhaust ports.
11. The power-output shaft is driven from the power turbine through a single-stage reduction gear.
12. The accessory-drive shaft is driven by the gasifier turbine through a set of reduction gears.

construction. The schematic in Fig. I-134 illustrates the air-flow paths and thermodynamic heat-transfer qualities of this heat exchanger.

GMC's GT-309 turbine engine, designed as an experimental unit for truck application, utilizes a horizontally mounted rotary drum-type heat exchanger. This unit, shown in Fig. I-135, is constructed of : metal matrix.

Figure I-136 illustrates the Pratt & Whitney rotary toroidal regenerator. This unit consists of a series of 24-mesh screen packs assembled into an annulus. Unit sealing is accomplished by a combination of bulkheads between the screen-pack sections and a sealing tunnel that separates the flow of cold and hot gas.

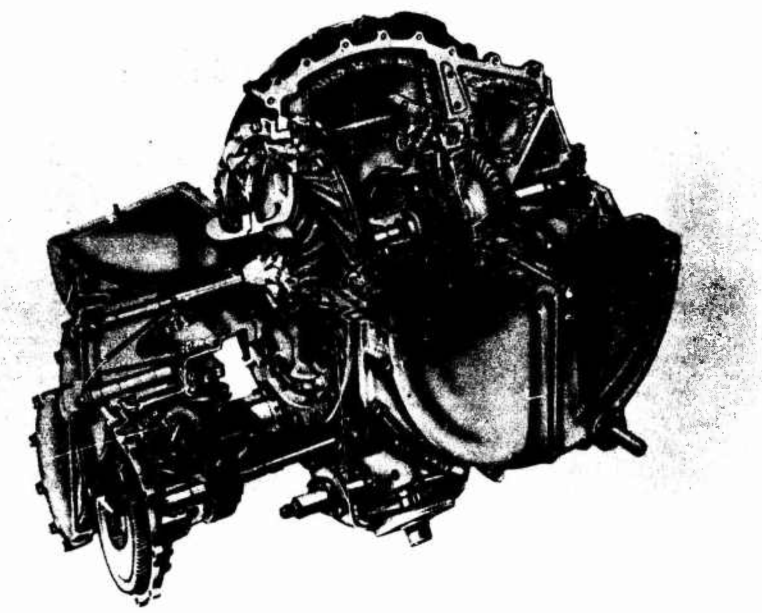


Fig. I-133—Cutaway View of Chrysler Corporation Model A-831 Turbine Engine, Showing Twin Vertical Rotating-Disk Metal-Matrix Heat Exchanger

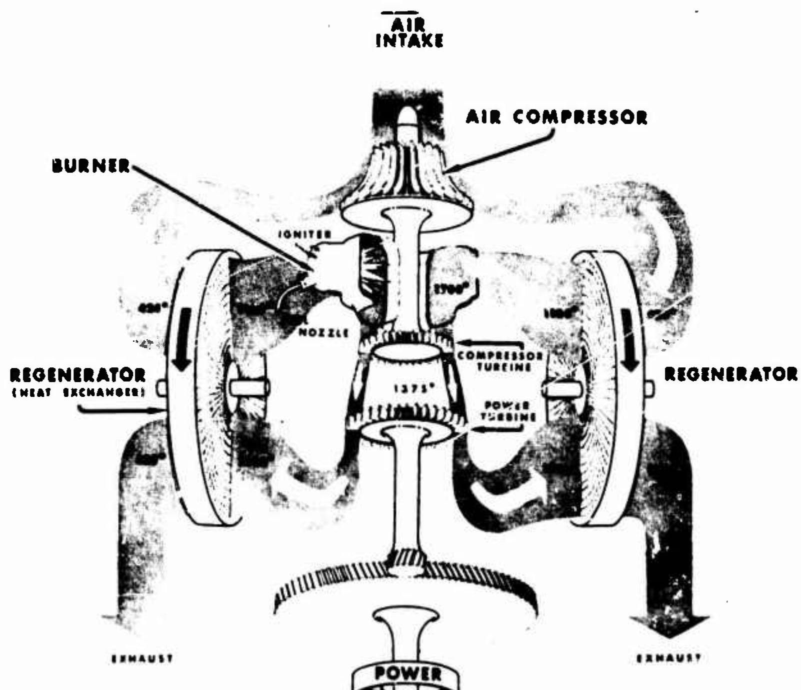


Fig. I-134—Schematic Flow Diagram of Chrysler Corporation Model A-831 Turbine Engine

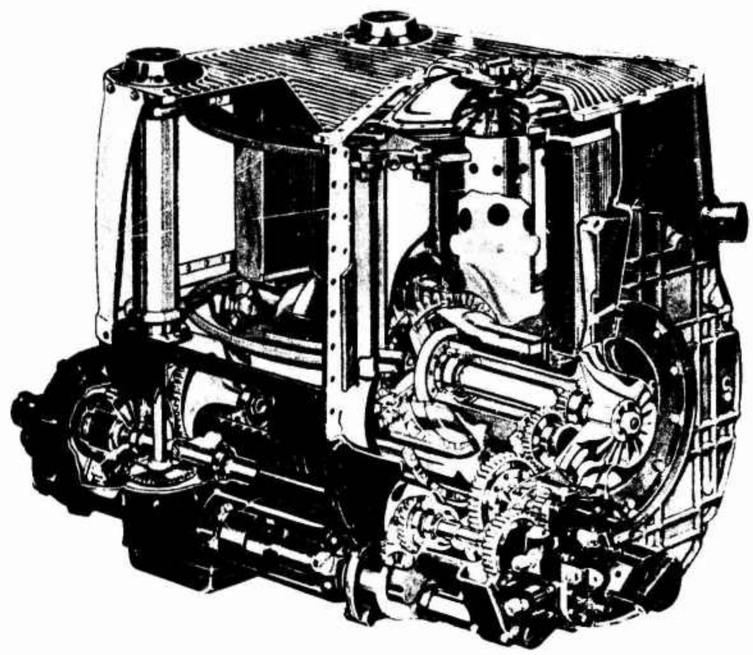


Fig. I-135—Cutaway View of GMC GT-309 Turbine Engine, Showing Horizontal Rotating-Drum Metal-Matrix Heat Exchanger

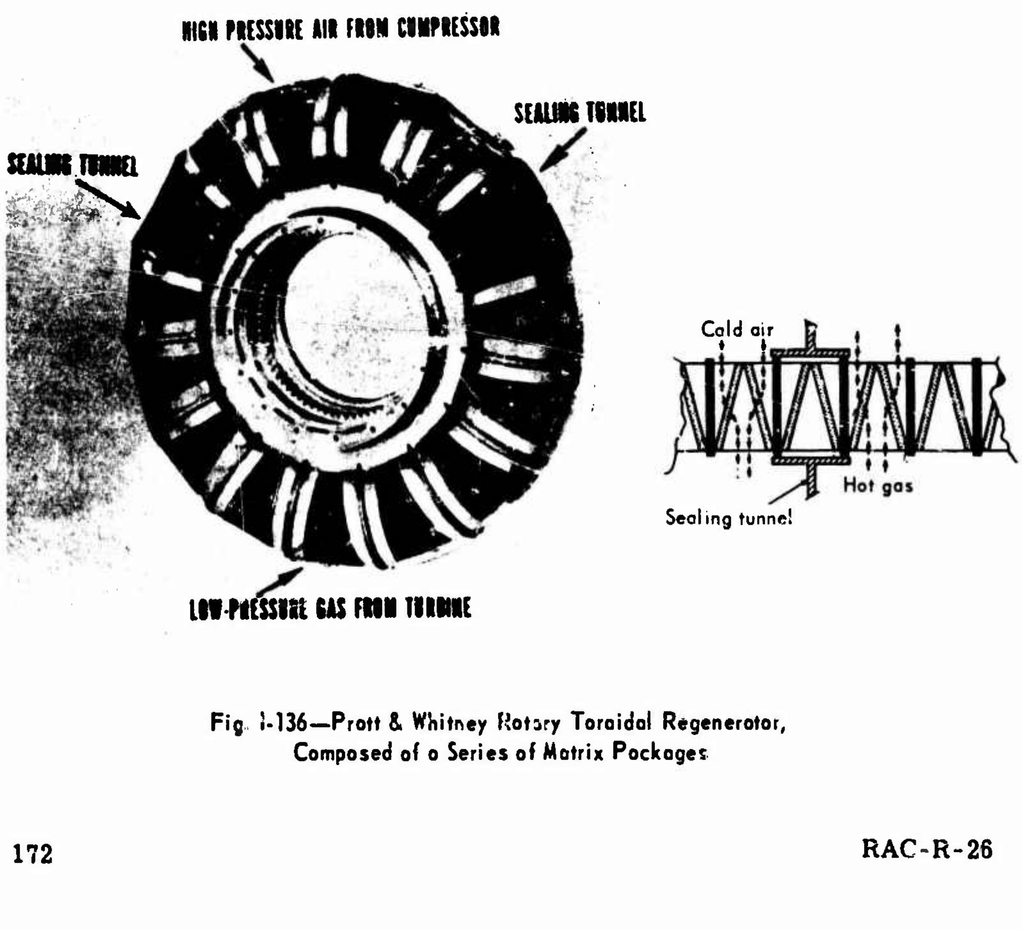


Fig. I-136—Pratt & Whitney Rotary Toroidal Regenerator, Composed of a Series of Matrix Packages

A recent development in heat-exchanger material has been made by the Corning Glass Works. The material is a thin-walled cellular glass-ceramic matrix. A cross section of a typical configuration is shown in Fig. I-137. A number of unique properties are combined in this structure, making it particularly suitable for use in rotary heat exchangers for regenerative gas-turbine engines. The material, trademarked "Cercor," has high temperature resistance, a nearly zero coefficient of expansion, extreme thermal-shock resistance, chemical durability, high specific-heat capability, and high strength at elevated temperatures. This material can be fabricated to a wide range of sizes in either the disk-type heat exchanger shown in Fig. I-138, or the drum-type heat exchanger shown in Fig. I-139.

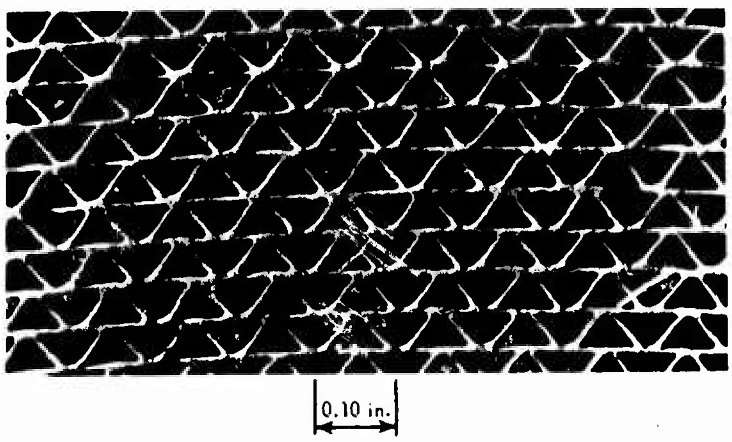


Fig. I-137—Typical Configuration (Enlarged View) of Corning Cercor Heat-Exchanger Structure

The Rover Company (England) was one of the first to utilize heat exchangers made of Cercor. Figure I-140 illustrates a cutaway view of the recent Rover 25/150R regenerative experimental automotive gas-turbine engine. The illustration shows the heat-exchanger mounting, drive scheme, and engine airflow paths. The ceramic heat exchanger is lightweight and compact, has greatest efficiency, and can be manufactured at a much lower cost than metal-matrix heat exchangers. Manufacturers of gas-turbine engines in the US are seriously pursuing the use of ceramic heat exchangers. The US automotive industries are designing the latest experimental automotive turbine engines to incorporate a ceramic heat exchanger. The Chrysler Corporation is considering modification of the Model A-831 turbine engine to incorporate a ceramic heat exchanger, and the Ford Motor Company has developed two new simple-cycle regenerative gas-turbine engines that utilize the ceramic heat exchanger. The new Ford engines are the 706, which develops approximately 180 to 200 hp, and the 707, which develops approximately 400 hp. Ceramic-core heat exchangers have demonstrated durability in automotive turbine-engine applications. However, a ceramic core may prove to be too brittle for application in military tactical-vehicle engines.

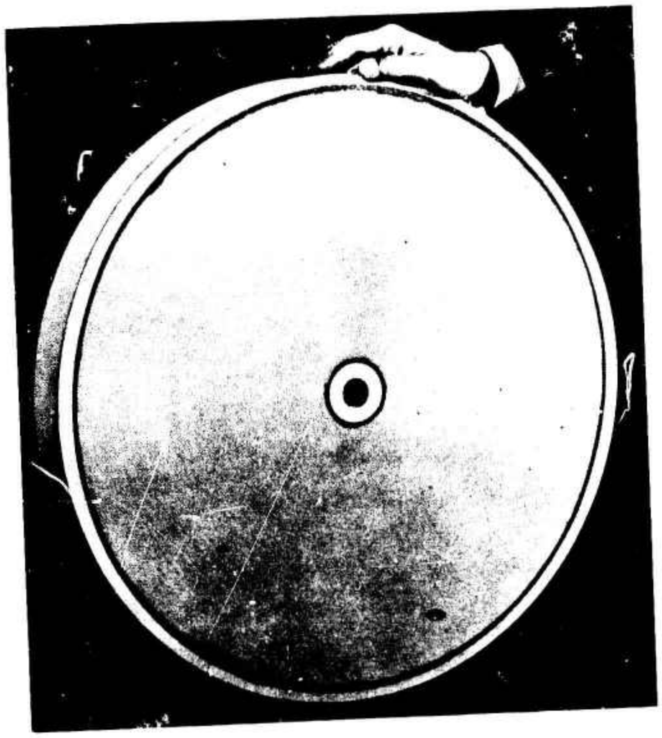


Fig. 1-138—Ceramic Disk-Type Rotary Heat Exchanger

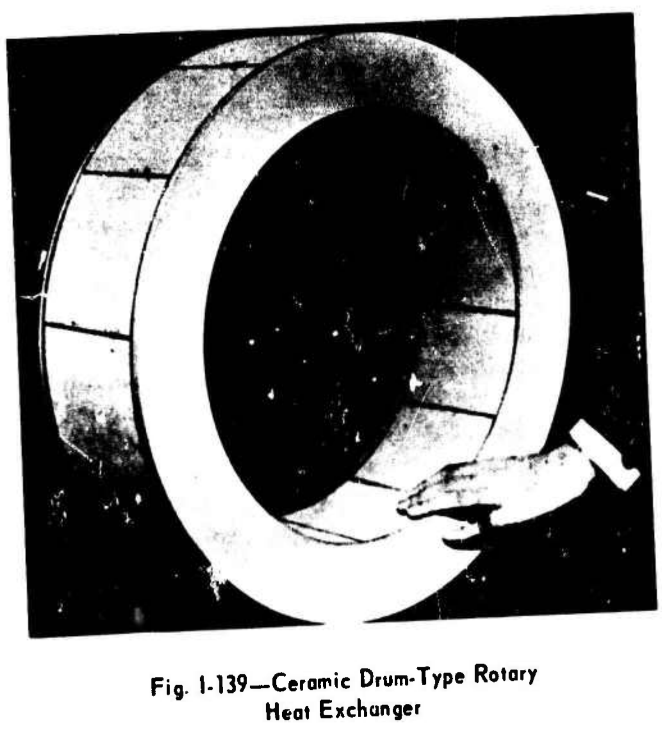


Fig. 1-139—Ceramic Drum-Type Rotary Heat Exchanger

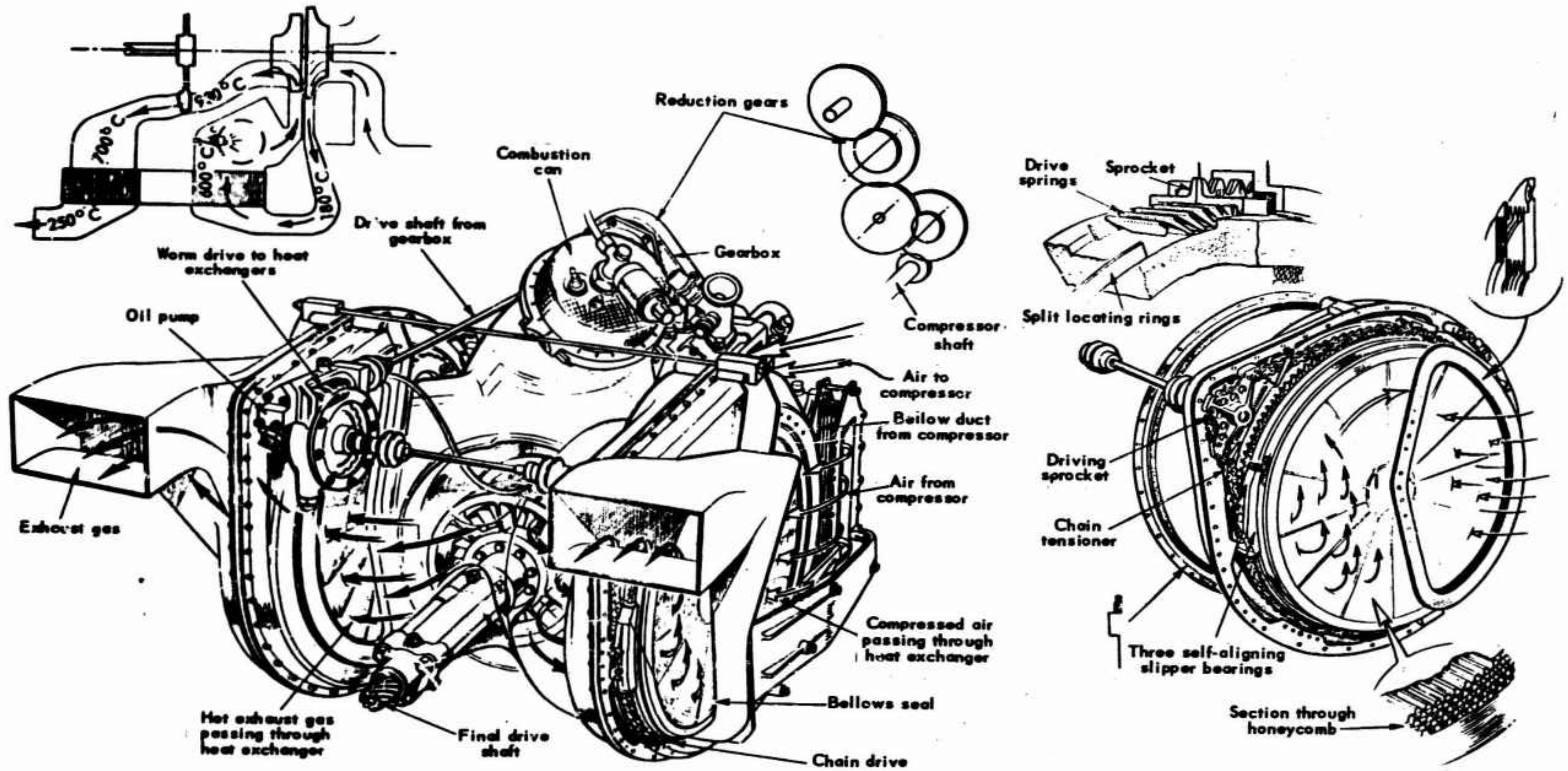


Fig. I-140—Cutaway View of Rover 25/150R Automotive Gas-Turbine Engine, Showing Vertically Mounted, Rotary Ceramic Disk-Type Heat Exchanger, and Air-Flow Paths

Side and rear air ducts removed from engine.

Stationary heat exchangers, commonly referred to as "recuperators," have been used in gas-turbine engines as heat-recovery devices. The stationary heat-exchanger designs are as varied as those of the rotary type. The main advantage of the stationary heat exchanger is the resultant elimination of mounting and drive hardware. Figure I-141 is a cutaway view of the Ford Model 705 turbocharged-cycle gas-turbine engine, which incorporates stationary metal-matrix intercoolers and recuperators. The Orenda OT-4 turbine engine, shown isometrically in Fig. I-142, utilizes twin stationary metal counterflow-type heat exchangers to recover exhaust-gas heat.

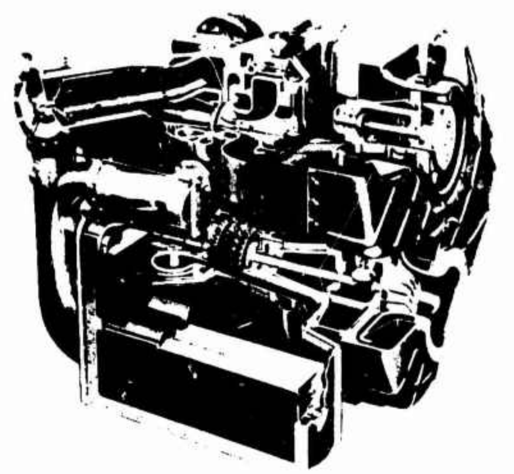


Fig. I-141—Cutaway View of Ford Model 705 Turbocharged-Cycle Turbine Engine, Showing Intercooler and Stationary Heat Exchanger

The Boeing Company developed a heat exchanger that was comprised of many metal tubes assembled into modules to form a stationary core. The concept of this heat exchanger is shown in Fig. I-143. One of the most common types of stationary heat exchangers used in gas-turbine engines is the direct-transfer tube type. This type of recuperator uses many thin-walled metal tubes assembled into a bundle and welded to end bulkheads. Figure I-144 illustrates a turbine engine incorporating a tube-type recuperator. The metal-tube recuperator is generally the least costly to build, although compared with the metal-plate or metal-matrix units, its efficiency is low. Heat exchangers that use liquid metals as the heat-transfer medium are generally constructed of tubes.

The AGT-1500 gas-turbine engine, under development by the Lycoming Division of AVCO Corporation for the new US Army main battle tank, utilizes a stationary heat exchanger. This unit is a stationary direct all-primary-surface convoluted-metal-plate heat exchanger. The design concept and air-flow paths are shown in Fig. I-145. For the most part the air and gas flow countercurrently to each other. Crossflow occurs only at the short air-entrance and exit regions. The convolutions on the plates are directed at right angles to

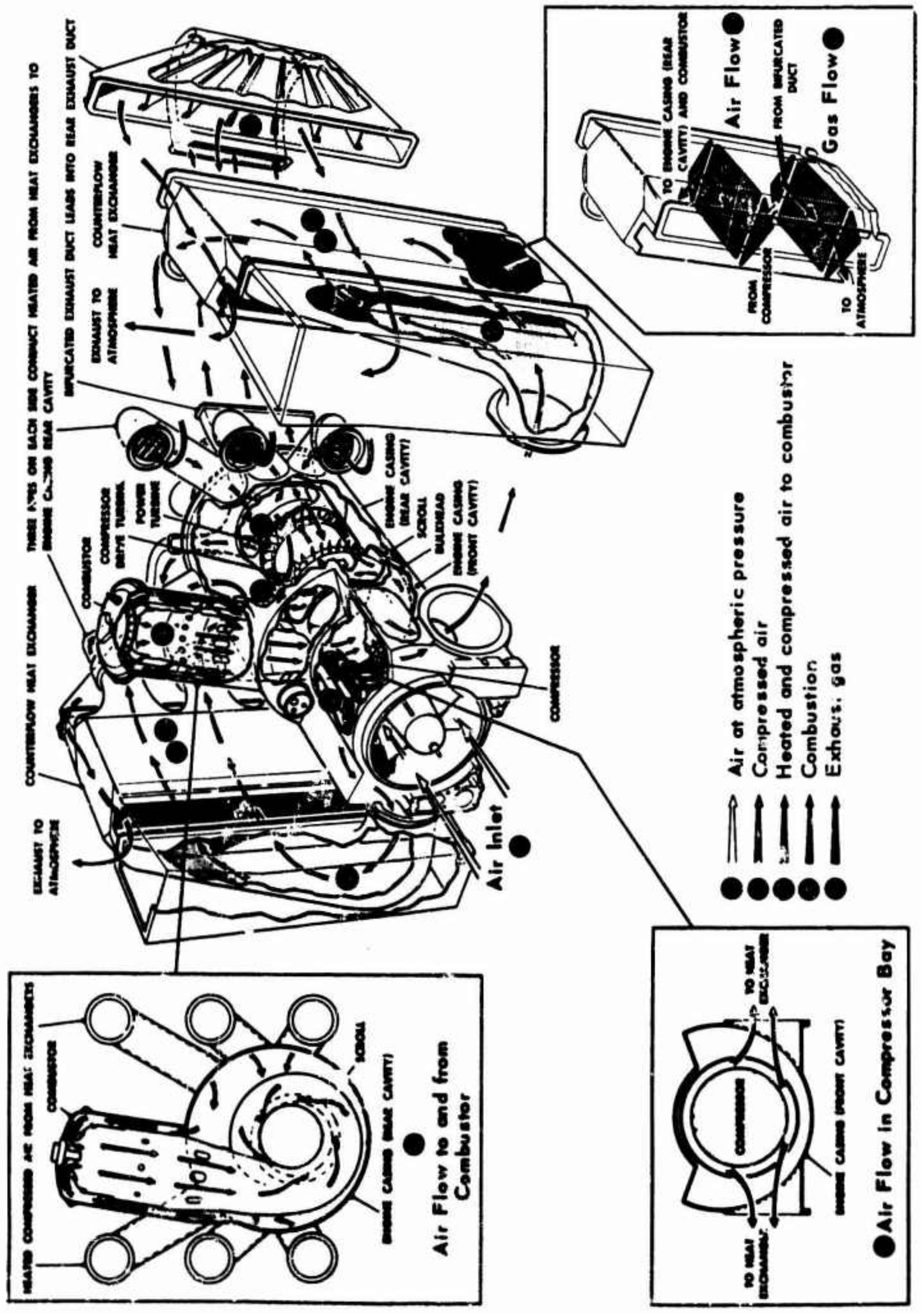


Fig. I-142—Isometric View of Orenda OT-4 Turbine Engine, Showing Stationary Metal Counterflow-Type Heat Exchanger

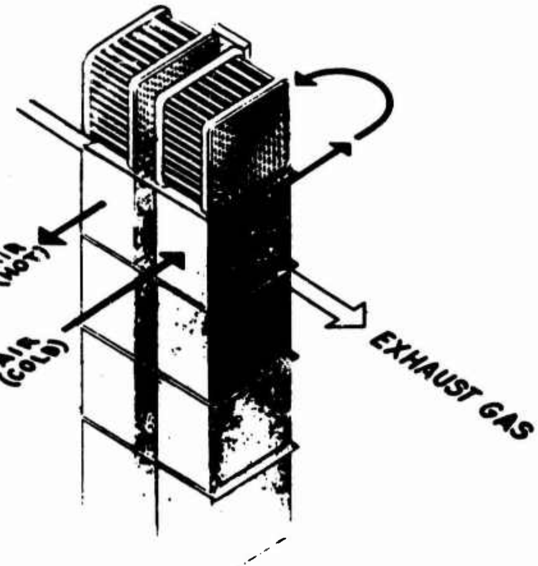


Fig. 1-143—Boeing Stationary-Core Heat Exchanger, Showing Individual Tube Modules

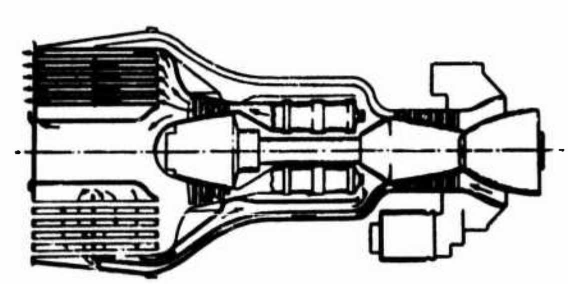


Fig. 1-144—Schematic Diagram of Gas-Turbine Engine Incorporating a Direct-Transfer Tube-Bundle Heat Exchanger

each other on alternate plates. In tests this type of regenerator has demonstrated a high degree of effectiveness.

Rotary heat exchangers are more efficient than stationary units. Existing rotary units have demonstrated an effectiveness of from 88 to 92 percent, whereas existing stationary units operate at an effectiveness of from 79 to 83 percent. The rotary heat exchanger can maintain a leakage rate over a longer period of time than a stationary heat exchanger but is more complex in that a drive mechanism and mounting system are required to allow rotation. Stationary heat exchangers are simpler in construction, easier to mount to the engine, and present less sealing problems. However, they are less effective.

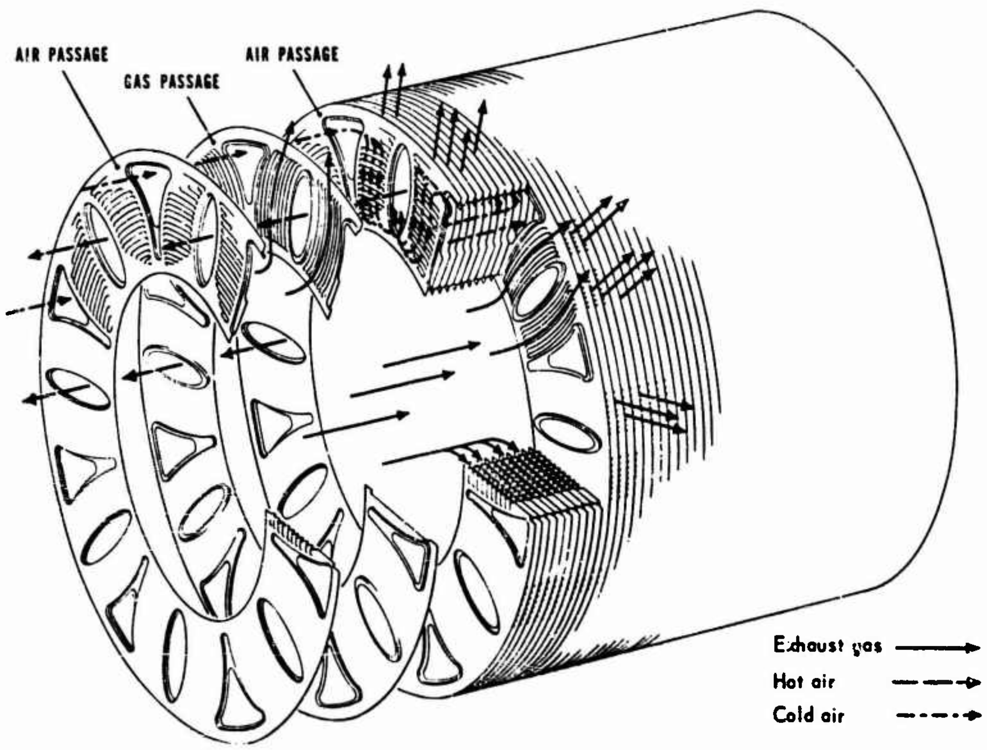


Fig. I-145—Concept and Flow Diagram of Lycoming Stationary Direct All Primary-Surface Convoluted-Metal-Plate Heat Exchanger

Heat exchangers are susceptible to fouling during engine operation on leaded fuels. To date a successful method of eliminating this problem (other than not using leaded fuels) has not been achieved. Heat exchangers are also susceptible to dust and dirt fouling. The rotary heat exchanger has a much greater self-cleaning capability than the stationary heat exchanger. The improvement forecast of heat-exchanger effectiveness is shown in Fig. I-146.

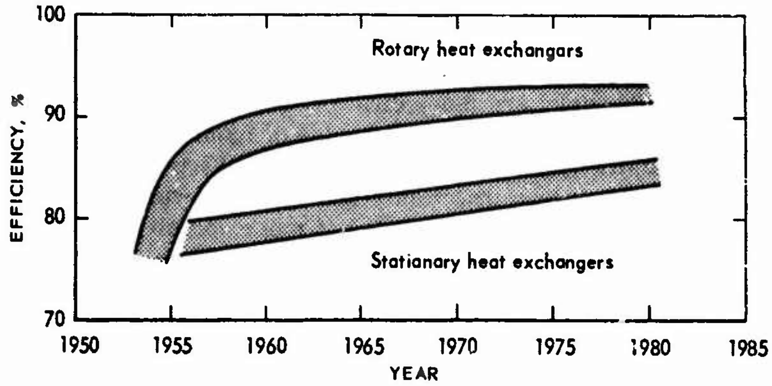


Fig. I-146—Improvement Forecast of Gas-Turbine Heat-Exchanger Effectiveness

ROTATING COMPONENTS

Significant progress has been achieved in the past few years in the design and efficiency of the rotating components in the gas-turbine engine. These components, i.e., compressors, gas-producer turbines, and power turbines, have not only become more efficient and rugged but are less costly to produce owing to advances in design technology and manufacturing techniques.

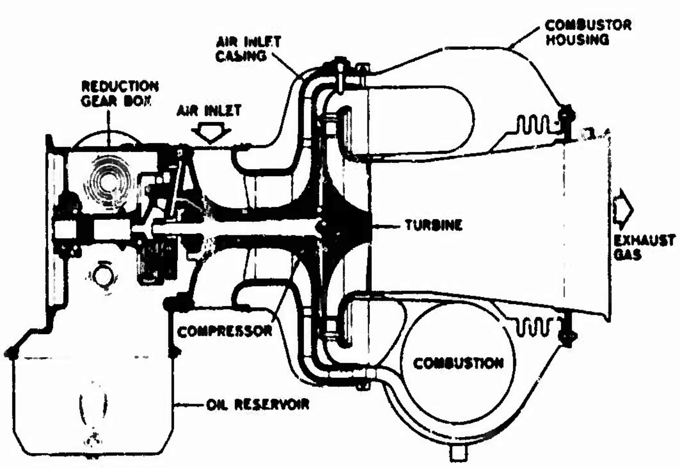


Fig. I-147—Small Single-Shaft Turbine Engine with Centrifugal Compressor and Radial-Inflow Gas-Producer Turbine

Gas-turbine compressors are of either centrifugal or axial design, or a combination of the two. Gas-producer turbines and power turbines are of either axial or radial-inflow design. Generally the smaller single-shaft turbine uses a single centrifugal compressor and a radial-inflow gas-producer turbine, as shown in Fig. I-147. The larger single-shaft engines incorporate multiple axial compressor stages (with possibly a centrifugal final stage) and axial gas-producer turbines. The smaller split-shaft turbine engines generally utilize a centrifugal compressor, axial gas-producer, and axial power-turbine wheels as do the engines shown in Figs. I-132, I-133, and I-135. Other combinations of compressors, gas-producer turbines, and power-turbine systems for turbine engines up to approximately 600 hp are shown in Figs. I-148 to I-150.

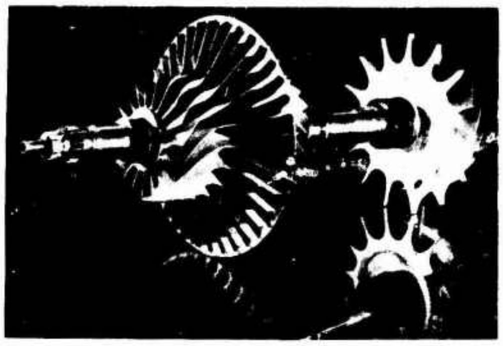


Fig. I-148—Single-Stage Double-Sided Centrifugal Compressor (foreground) and Radial-Inflow Gas-Producer Turbine (right)

This high-performance compressor delivers air at 6.. atmospheres.

Fig. I-149—Gas-Producer Section of 600-hp Turbine Engine with Two Centrifugal Compressor Stages and Three Axial Gas-Producer-Turbine Stages

Large gas-turbine engines, like the one shown in Fig. I-151, utilize a multistage axial compressor with a final centrifugal stage, a single-stage axial gas-producer turbine, and a two-stage axial power turbine. The US Army AGT-1500 gas-turbine engine, shown in Fig. I-152, uses a low-pressure and high-pressure multistage axial compressor, a high- and low-pressure axial gas-producer turbine, and a two-stage axial power turbine. Figure I-153 illustrates a high-efficiency (84 to 85.5 percent) first-stage supersonic compressor that is being developed by several companies under an Army contract for the development of advanced-technology components for small gas-turbine engines.

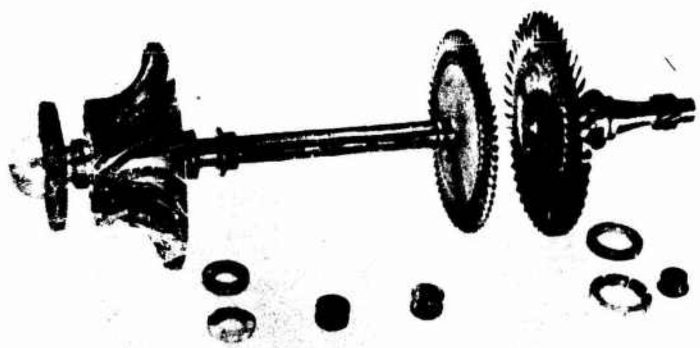


Fig. I-150—Gas-Producer and Power Section of 400-hp Turbine Engine with Axial-Centrifugal Compressor (left), and Axial Gas-Producer Turbine and Power Turbine

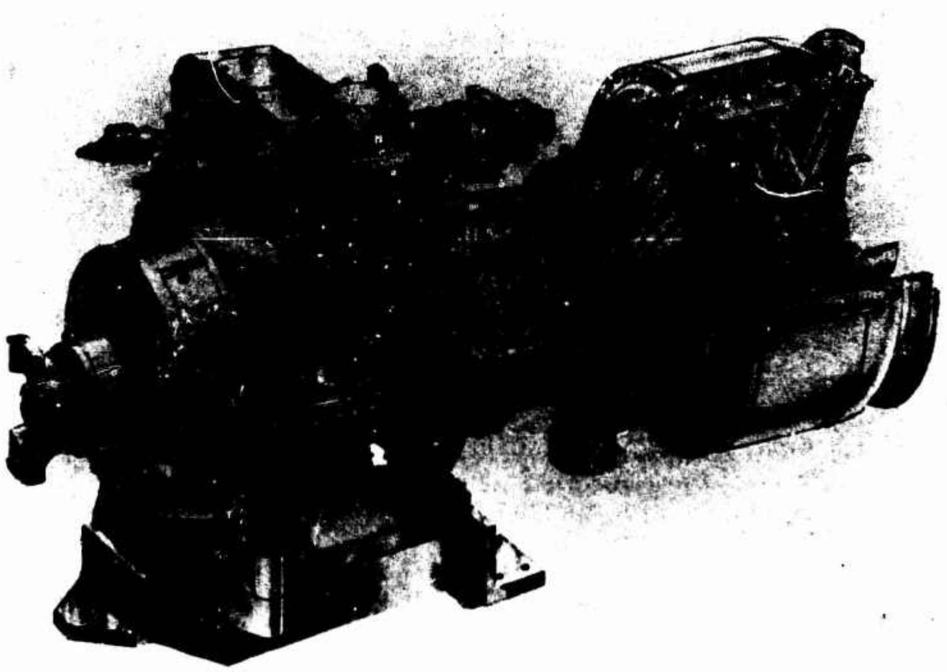


Fig. I-151—Cross Section of 1500-hp Unregenerated Marine Turbine Engine (Lycoming TF-20) with Axial-Centrifugal Compressor, Single-Stage Axial Gas-Producer Turbine, and Two-Stage Power Turbine

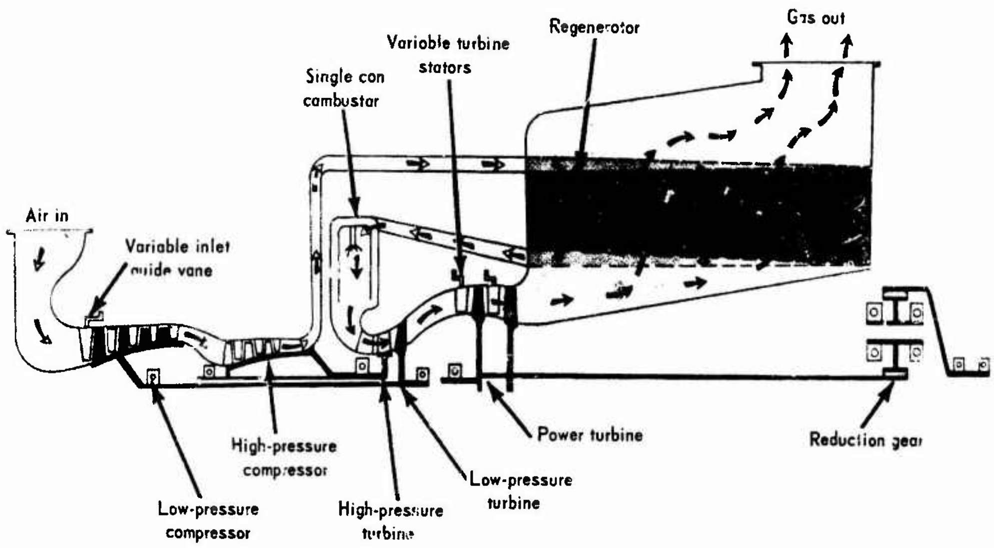


Fig. I-152—Schematic Cross Section of US Army AGT-1500 (Lycoming PLT 25) Future Main Battle Tank Turbine Engine, with Multistage Compressor, Gas-Producer Turbine, and Power Turbine

This engine incorporates variable inlet-guide vanes and variable turbine stators.

Fig. I-153—First-Stage Supersonic Compressor

Characteristic	Goals	
	Mod 1	Mod 2
Pressure ratio	2.0	2.8
Airflow, lb/sec	4.0	4.0
Efficiency, %	84.3	85.5
Tip speed, ft/sec	1395	1395
Rotor-inlet Mach No.	1.51	1.51

The fundamental factors that determine the performance of gas-turbine engines are the type of cycle, engine aerodynamic efficiency, and turbine temperature. The fuel-consumption characteristics as a function of pressure ratio of various regenerative cycles are shown in Fig. I-154. As shown, the optimum shaft specific fuel consumption (SSFC) of the regenerative cycle is only some 7 percent less than that of a Brayton cycle with a pressure ratio of 12 to 1 for the assumed conditions. It is generally true that this SSFC difference is not very high at the design conditions shown. But improvements of the order of 30 to 40 percent can be obtained at low power settings. Also, as shown, the regenerative cycle with intercooling achieves SSFC some 8 percent lower than

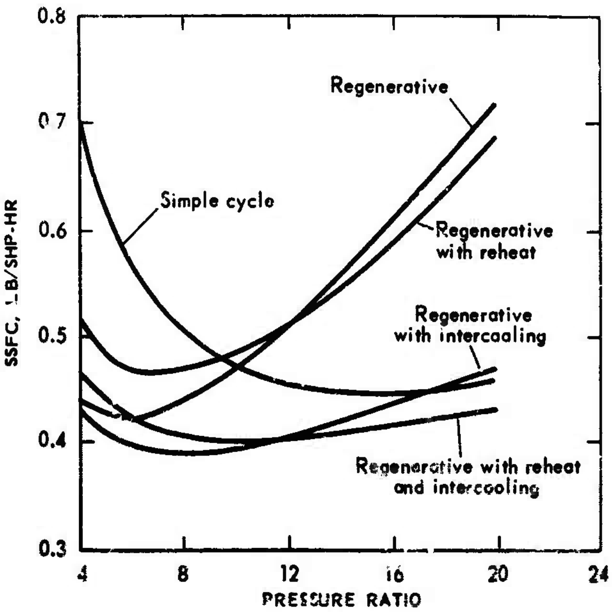


Fig. I-154—Comparison of SSFC of Regenerative Cycles
Turbine temperature 1600° F,
Reheat temperature 1400° F,
Intercooler effectiveness 70%,
Regenerator effectiveness 80%.

the simple regenerative cycle and a specific power that is 20 percent higher. Reheat causes the SSFC of the regenerative cycle to increase through the range of present pressure ratios but also causes an important increase in specific power that can conceivably be used for intermittent increases in power. Reheat and intercooling, as incorporated in several industrial and one automotive gas turbine, cause some improvement in the SSFC of the regenerative cycle, but, most importantly, cause the specific power to increase 40 percent.[1]

Figure I-155 illustrates the performance goals of advanced-design compressors being developed by the US Army. The curves illustrate that a small gain in pressure ratio and efficiency results in a significant increase in power and a decrease in fuel consumption. Figures I-156 and I-157 illustrate specific power and SSFC as a function of pressure ratio with varying turbine-inlet temperatures. The graph illustrates the performance of current engines and

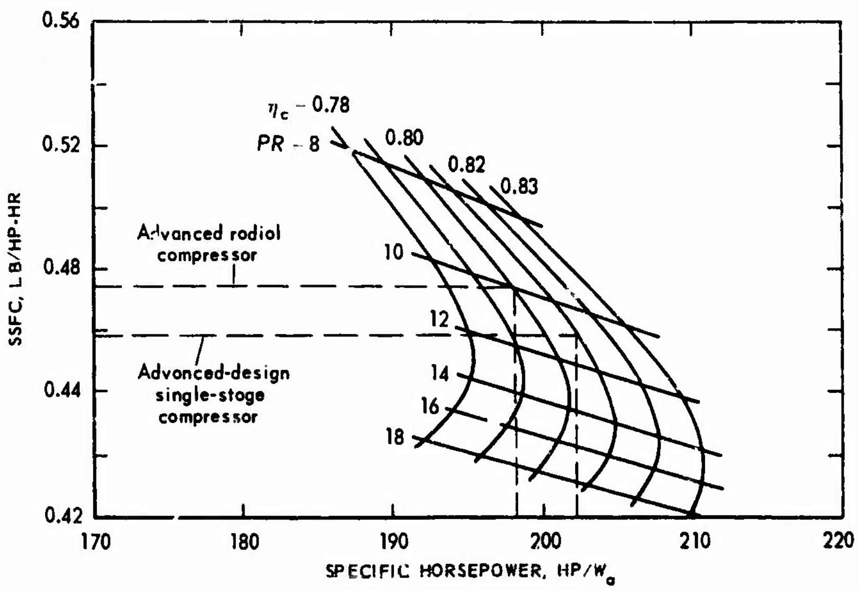


Fig. I-155—Effect of Compressor Characteristics on Engine Performance

Turbine-inlet temperature - 2300° F
$\eta_{gas\ turbine}$ - 89%
$\eta_{pressure\ turbine}$ - 89.5%
η_{nozzle} - 95%
Mech. Eff. pressure turbine - 98%
BLEED - 5%

the objectives for future engines, based on the advanced-component-technology program of the Army. Figure I-156 indicates a significant increase in power due to increased pressure ratio and turbine-inlet temperature. Figure I-157 indicates that gas-turbine engines operating with pressure ratios of 12 to 1 and turbine-inlet temperatures of 2600°F will have significantly greater fuel economy than today's engines. Present simple-cycle gas-turbine engines operate at pressure ratios of 3.5 to 6:1. The Ford experimental model 705 turbocharged-cycle engine operates at a pressure ratio of 16 to 1. Advanced-technology simple-cycle turbine engines now under development will operate at pressure ratios of 10:1 to 12:1 and have an ultimate goal of 16 to 1. Engine operation at high pressure ratios is necessary to reduce the amount of intake air required by the turbine, thereby reducing the air-cleaner and heat-exchanger size requirements.

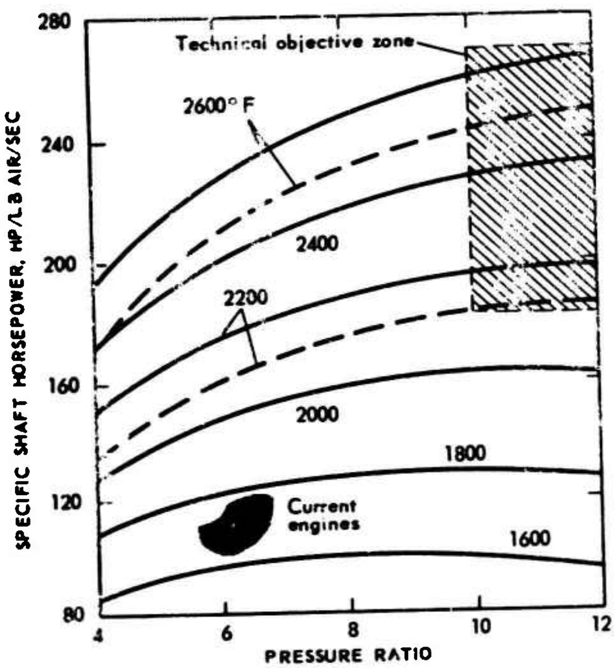


Fig. I-156—Specific Power as a Function of Pressure Ratio with Varying Turbine-Inlet Temperatures

— — — With regeneration

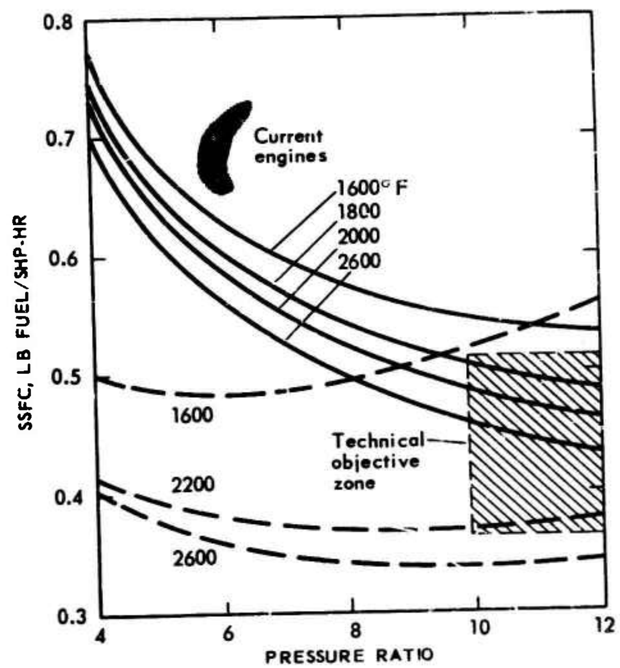


Fig. I-157—SSFC as a Function of Pressure Ratio with Varying Turbine-Inlet Temperatures

——— No generator — — With regeneration

The improvement forecasts of compressor, gas-producer-turbine, and power-turbine effectiveness are shown in Figs. I-158 to I-160.

Gas-turbine engines have a very high level of combustion efficiency. Combustion efficiencies of 98 to 99 percent have been achieved. Because of the nearly complete combustion, the exhaust of the gas-turbine engine contains only a negligible amount of toxic carbon monoxide. Gas-turbine engines incorporate a combustor, or burner, design that is either annular or can-shaped.

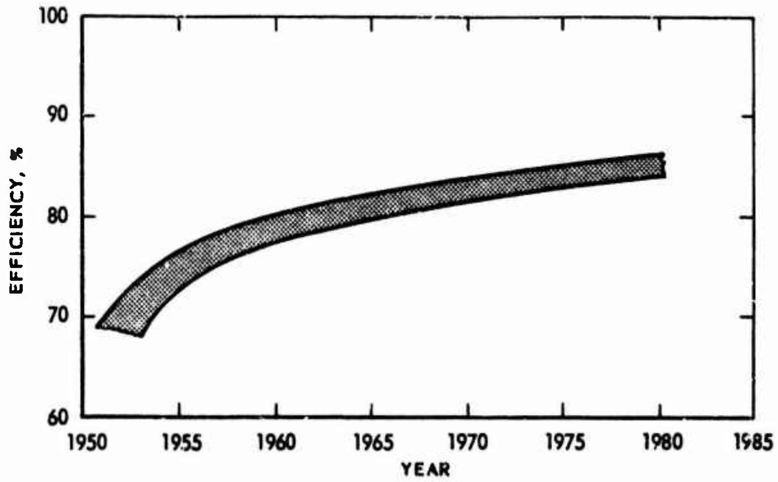


Fig. I-158—Improvement Forecast of Compressor Effectiveness

Both designs have attained similar high efficiencies. The engine shown in Fig. I-151 incorporates an annular combustor. Improved combustor metals and proper cooling design will be necessary for engine operation at high temperatures. Before turbine-inlet temperatures can increase, combustor operating temperatures must increase. Proper combustor design must have good dirt-swallowing capability and a minimum surface for a given heat release to ensure proper cooling of the combustor walls. Atomizing can combustors have been operated at approximately 3100°F, and annular atomizing combustors have operated at approximately 2500°F under laboratory conditions. Present combustors operate at 1800 to 2100°F. The improvement forecast of combustor effectiveness is shown in Fig. I-161.

To achieve high performance improvements for increased power output and to reduce fuel consumption, turbine operating temperature must be increased. The effect of turbine-inlet temperature and pressure ratio on fuel consumption and power output has been shown. The performance gains due to increased turbine-inlet temperature are significant. Figure I-162 illustrates the decrease in fuel consumption due to increased turbine-inlet temperature. The increase in power output per pound of air due to increased turbine-inlet temperature is shown in Fig. I-163.

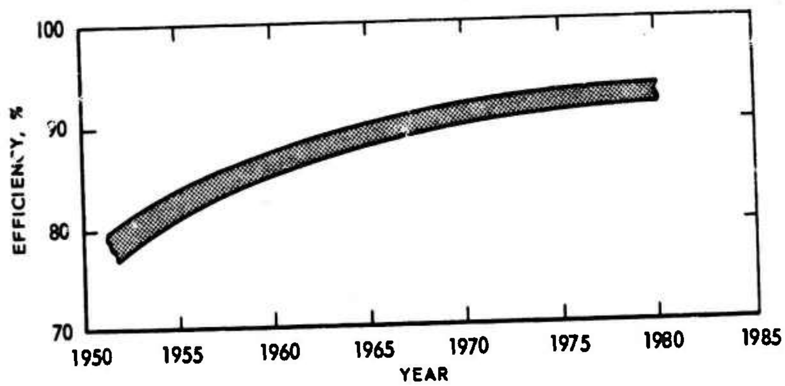


Fig. I-159—Improvement Forecast of Gas-Producer-Turbine Effectiveness

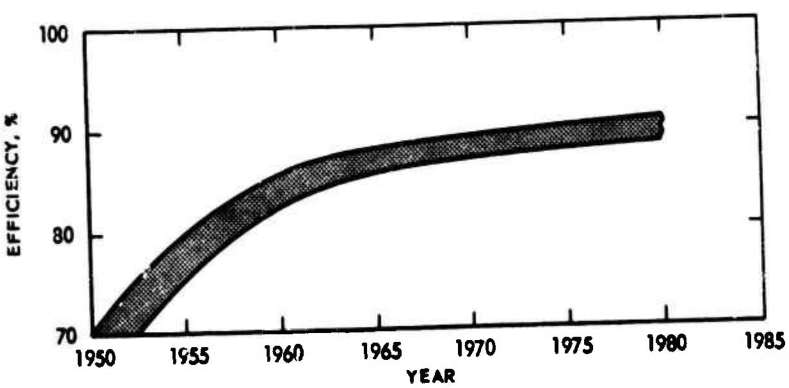


Fig. I-160—Improvement Forecast of Power-Turbine Effectiveness

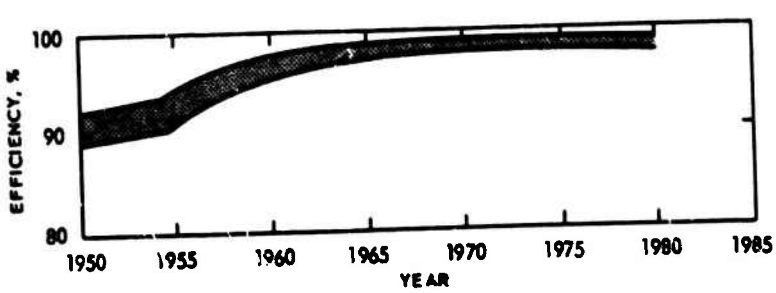


Fig. I-161—Improvement Forecast of Burner Effectiveness

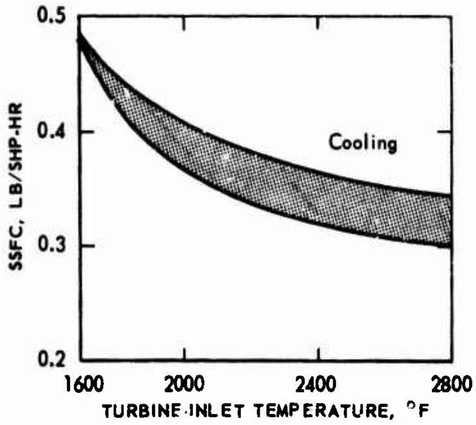


Fig. I-162—SSFC Performance Gains with Increased Turbine-Inlet Temperature

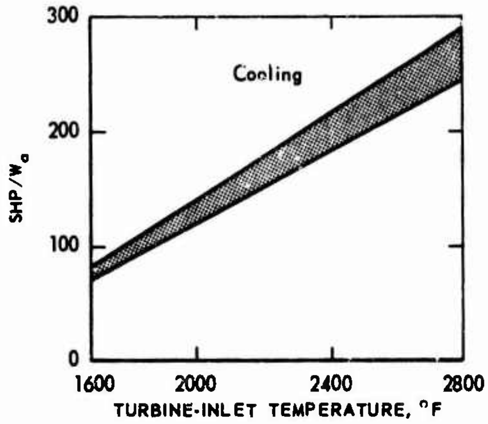


Fig. I-163—Power-Output Performance Gains with Increased Turbine-Inlet Temperature

COOLING METHODS

At gas temperatures above 1700 to 1800°F, blade cooling is necessary to keep metal temperatures within limits of current technology. Improvements in turbine-blade-metal creep strength (expressed as the increase in temperature above a datum value at which the material can be operated to give the same creep life at the same stress level) due to metallurgical improvements have increased an average of 18 to 20°F per year since 1951.[2] Turbine engines of a few years past incorporated uncooled turbine blades of either solid or hollow construction. Much research work has been concentrated on the cooling of turbine blades to achieve high-performance operation. Blade cooling in large aircraft turbines has advanced rapidly in the past 5 years. The cooling of turbine blades in aircraft engines, however, is easily accomplished because of the comparatively large blading and air-channeling components compared to the small size of these components in surface-vehicle engines. The advent of of the integral cast turbine blade-wheel assembly has made cooling much more difficult in engines designed for surface vehicles.

The performance characteristics of the various turbine blade-cooling concepts are shown in Fig. I-164, which gives the relation of percentage of cooling airflow requirements to turbine-inlet temperature. Increased cooling airflow, as a percentage of total engine air requirements, results in decreased fuel efficiency and a reduction of power output. Of the four blade-cooling concepts—convection, impingement, film, and transpiration—the latter three concepts offer the greatest potential for increased operating temperatures with minimum cooling-air requirements. Figure I-165 illustrates the evolution of various air-cooled turbine-blade configurations in order of increasing effectiveness.

Convection-cooled cast turbine blades have demonstrated reliable operation at 2000 to 2300°F. Figure I-166 illustrates the principle of convection cooling of turbine blades. Relatively cool turbine bleed air passes through an

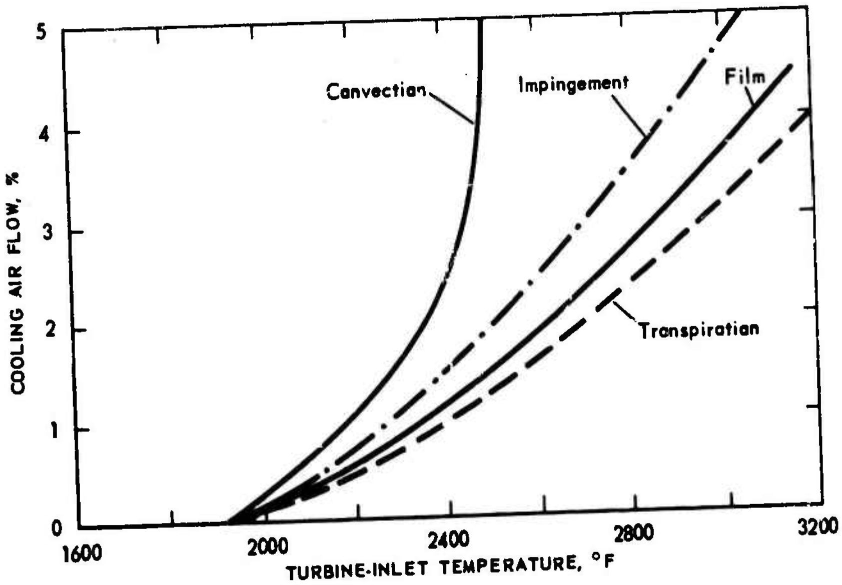


Fig. I-164—Comparative Cooling-Air Requirements for Rotor Blades, by Type of Cooling Concept

Comparative cooling requirement for rotor blade, Rc = 4 : 1; leading edge metal temperature = 1800° F.

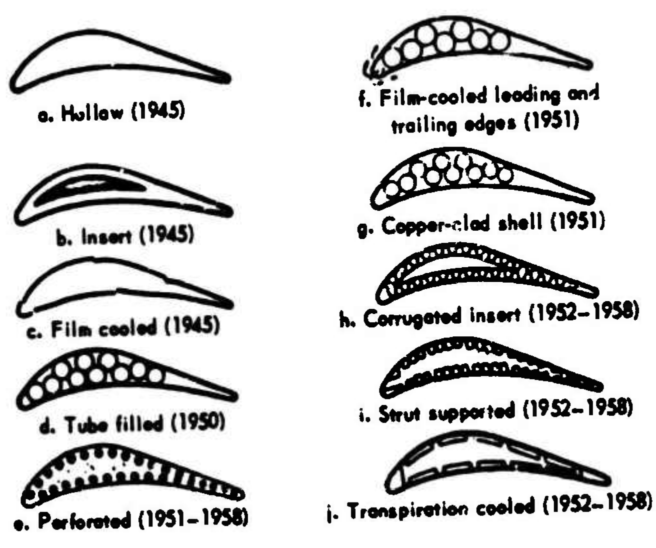


Fig. I-165—Evolution of Cast Internally Cooled Turbine Airfoils

interval passage or passages in the blade. Advanced-design convection-cooled blades, such as those shown in Fig. I-167, have demonstrated operation in excess of 2700°F in laboratory test rigs. Figure I-168 illustrates cross sections of typical single-, double-, and triple-pass convection-air-cooled turbine blades used in Rolls-Royce aircraft engines. These blade designs operate at turbine-inlet temperatures of 1900 to 2100°F and have service lives of 5000 to 10,000 hr of operation at these temperatures. One manufacturer has recently developed a technique for casting a monolithic wheel and blading for small turbine engines that has an internal air plenum and internally finned and passaged blades. This monolithic wheel requires a minimum of machining. The total cost of this wheel-and-blade assembly has been reduced to $800 from $2400 for a wheel approximately 8 in. in diameter.

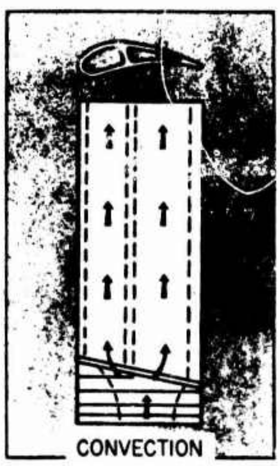


Fig. I-166—Convection-Cooled Turbine Blade
Cooling air passes through internal passages in blade.

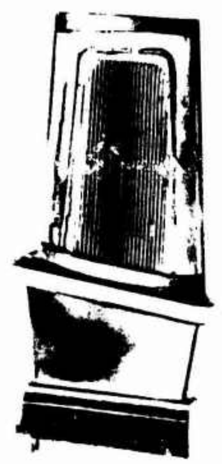


Fig. I-167—Advanced-Design Convection-Cooled Turbine Blade with Internal Fins

Impingement cooling involves the use of a fabricated tube inserted into a hollow vane. The cooling air is jetted at the inner wall of the blade leading-edge surface and is exhausted chordwise through holes in the vane wall ahead of the pressure side of the blade. This system, shown in Fig. I-169, is best applied to the cooling of first-stage stator vanes.

Film cooling of turbine and stator blades is another effective means of attaining high turbine-inlet temperatures. Film-cooled turbine blades, as shown in Figs. I-170 and I-171, have been operated in excess of 2600°F. All-film-cooled turbine blades, as shown in Fig. I-172, offer advanced cooling concepts with a potential of operating at turbine-inlet temperatures of 2800 to 3000°F.

Transpiration cooling appears to offer a means of obtaining maximum cooling effectiveness. The principle of transpiration cooling is shown in Fig. I-173. Cooling air passes through a porous wall in the airfoil, cooling the blade metal. The air then forms a film of relatively cool air that insulates the outer surface of the airfoil from the stream of hot gas. The transpiration fabric is

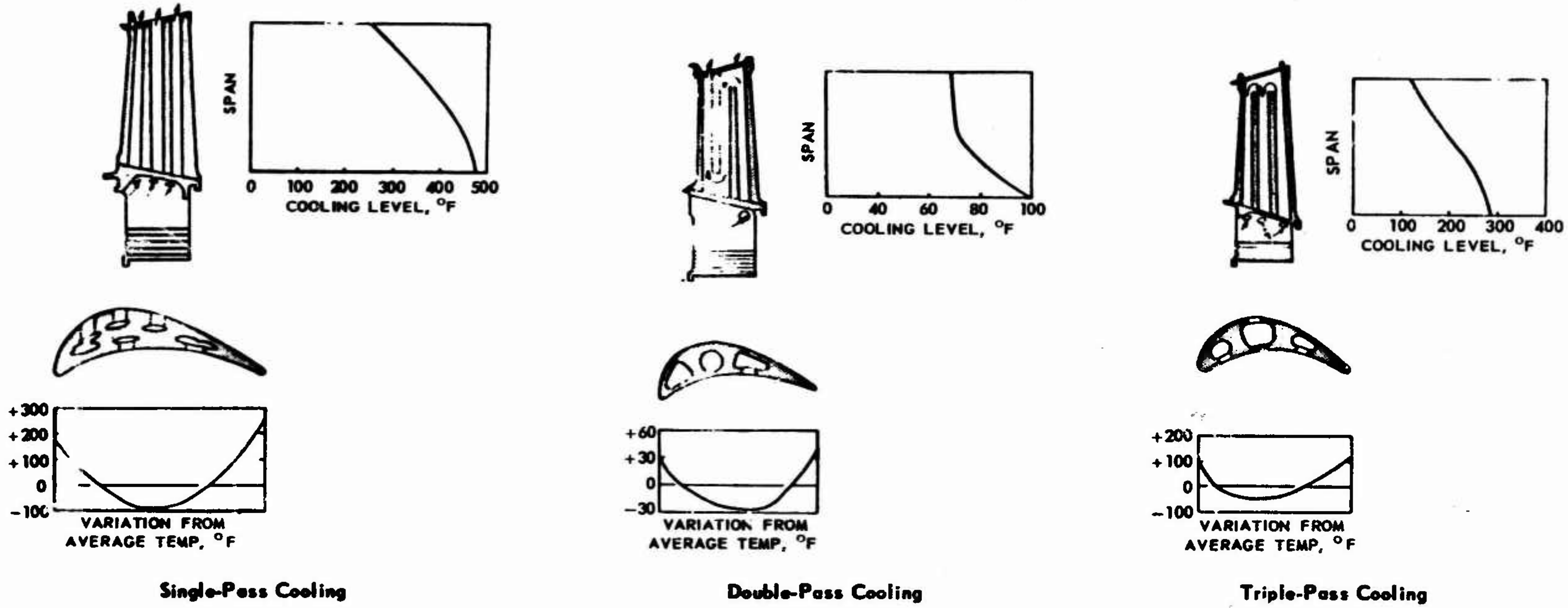


Fig. I-168—Typical Convection-Air-Cooled Aircraft-Engine Turbine Blades (Rolls-Royce)

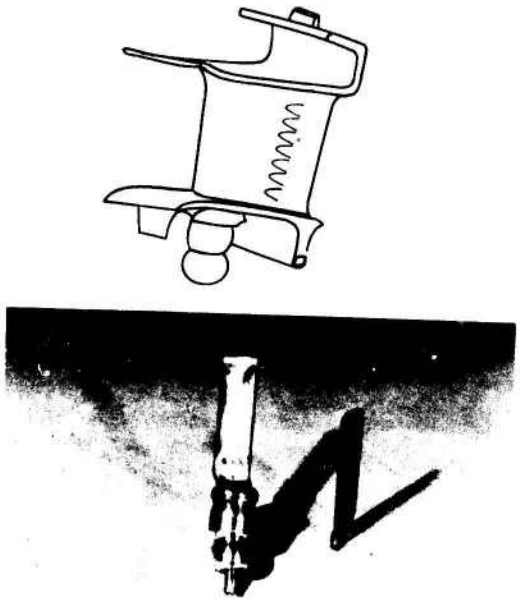


Fig. I-169—Impingement-Cooled First-Stage Stator-Vane Assembly (Allison)

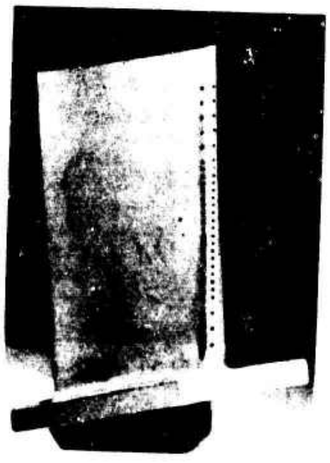


Fig. I-170—Leading-Edge Film-Cooled Blade

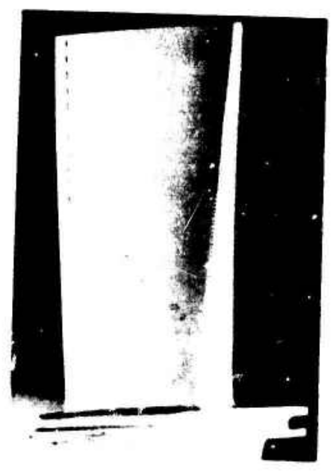


Fig. I-171—Film-Convection-Cooled Blade with Leading and Trailing Edge Film-Cooled

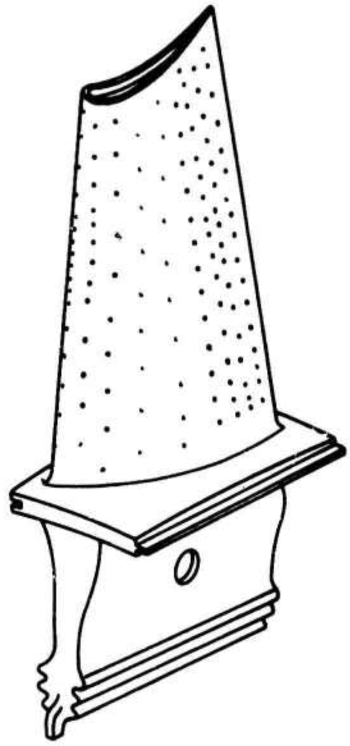


Fig. I-172—Typical All-Film-Cooled Turbine Blade

Fig. I-173—Transpiration-Cooled Turbine Blade

Cooling air passes through pores in airfoil, cooling the blade metal, and then forms a film of relatively cool air that insulates the outside of the blade surface from the stream of hot gas.

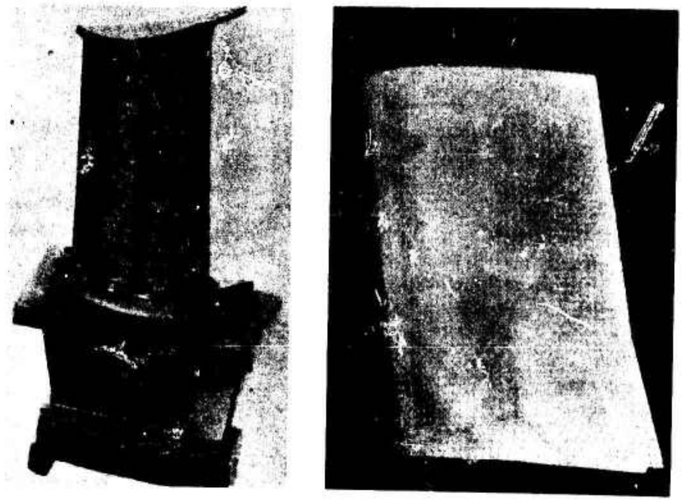


Fig. I-174—Cast Support Strut and Porous Shell Prior to Joining Assembly
(Allison)

Fig. I-175—Transpiration-Cooled Turbine-Blade Assembly
Incorporating Lamilloy Airfoil
(Allison)

made from thin sheets of sintered and/or pressed-wire-type laminates, stacked and diffusion-bonded. These materials have poor structural integrity and must be supported by a rigid strut. The Allison Division of the GMC has developed transpiration materials called Porolloy, Lamilloy, and Rigimesh that have demonstrated excellent cooling characteristics. Turbine blades using this material, strut supported, are illustrated in Figs. I-174 to I-176. Figure I-177 illustrates

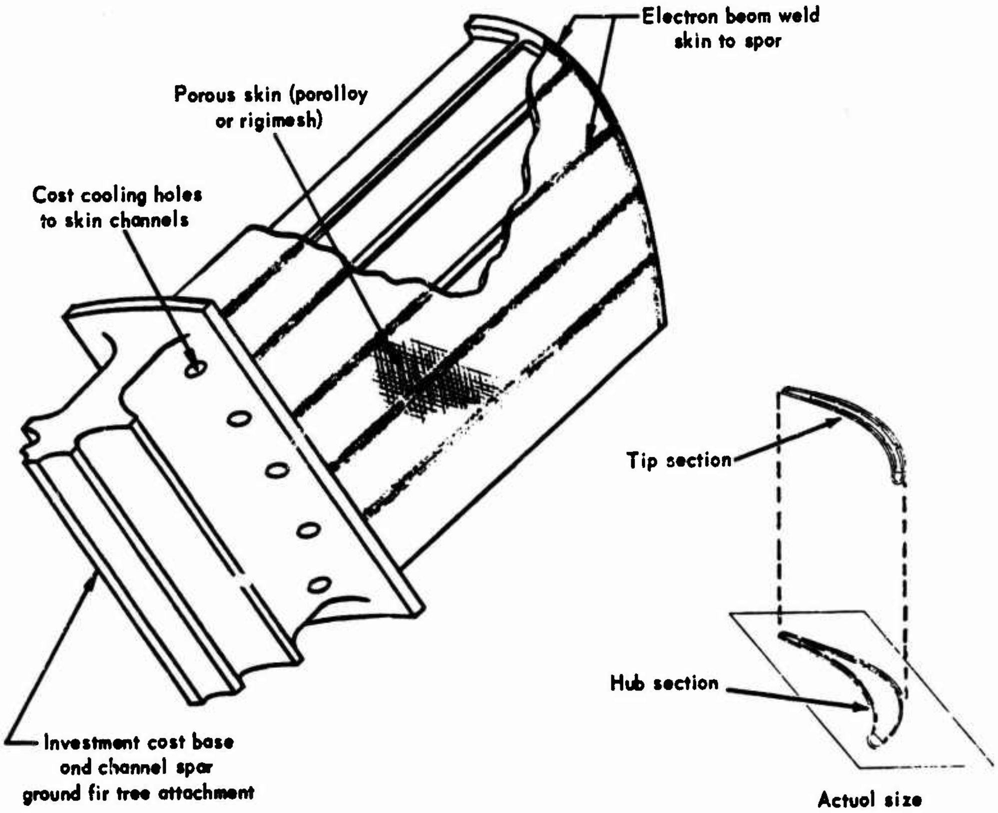


Fig. I-176—Strut-Supported Transpiration-Cooled Turbine Blade

a transpiration-cooled stator blade designed to operate at 2500°F. One manufacturer has developed an advanced-design transpiration-cooled variable-permeability self-supporting blade that has demonstrated high-temperature operation. The feature of this blade material, aside from high-temperature-operation capabilities, is that turbine blades can be manufactured from it at low cost. Transpiration cooling of turbine blades offers the potential of operating at turbine-inlet temperatures of 2800 to 3200°F with cooling-air flows of 4 percent of total engine air flow. Transpiration-cooled airfoils have demonstrated cooling capabilities of 2500 to 2700°F in laboratory test rigs. Liquid cooling of turbine blades by either water or fuel looks promising, and much

Fig. I-177—Enlarged Section of Transpiration-Cooled Stator-Blade Assembly Designed To Operate at Gas Temperatures of 2500° F
(Curtiss-Wright)

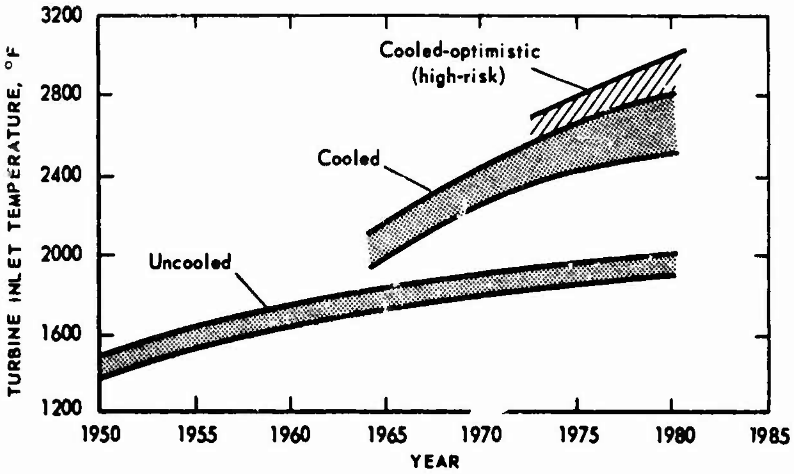


Fig. I-178—Improvement Forecast for Turbine-Inlet Temperature

work is being done in this area. The improvement forecast of increased turbine-inlet operating temperature for uncooled, cooled, and cooled-optimistic goals, is shown in Fig. I-178. Figure I-179 illustrates engine cycle performance for SSFC as a function of specific horsepower, as affected by increased pressure ratio, regenerator effectiveness, and turbine-inlet temperature.

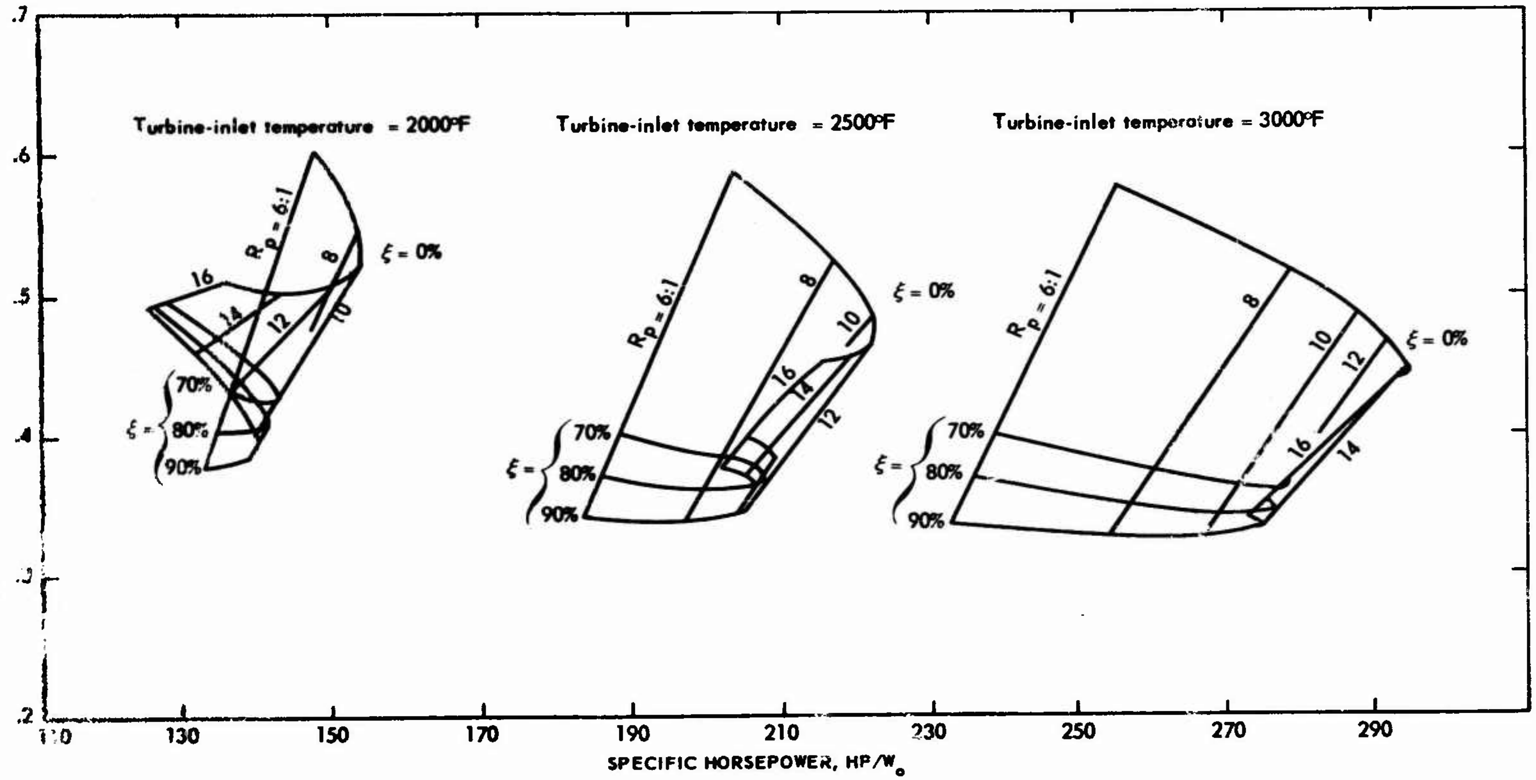


Fig. I-179—Cycle Performance, SSFC as a Function of Specific Horsepower

Showing effect of increased pressure ratio, regenerator effectiveness, and turbine-inlet temperature.

Assumptions

Regen. Press. Drop – % = $\frac{\xi\%}{10}$

Turb. Cooling Bleed

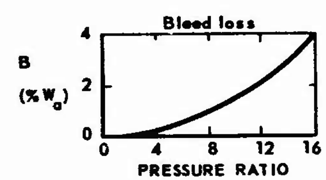
$\beta_c - \%\, W_o = 2.3 + \frac{\text{Turbine-inlet temperature °F} - 1800}{200}$

η_c = 80%

η_{gt} = 86%

η_{pt} = 89.5%

η_{comb} = 98.5%



VARIABLE-GEOMETRY NOZZLES

To achieve low SSFC and adequate dynamic braking of the power turbine section of the engine at part loads, the use of variable-geometry nozzles is required. The use of variable-geometry nozzles at the power-turbine inlet enables the engine to maintain a high turbine-inlet temperature at part loads. Maintaining a high cycle temperature at part-load conditions is necessary for low fuel consumption. The mechanically independent free turbine must incorporate variable-geometry nozzles to provide maximum overall system performance for all operating conditions. The Chrysler Corporation Model A-831 automotive gas-turbine engine incorporates a variable-nozzle system, as shown in Fig. I-180. The nozzles are varied automatically by speed- and power-demand-sensing units on the engine. The four nozzle positions are idle, modulating range, full power, and braking position (retardation). Automobile braking characteristics with this system are comparable to those of a conventional piston engine with torque converter, as shown in Fig. I-181.

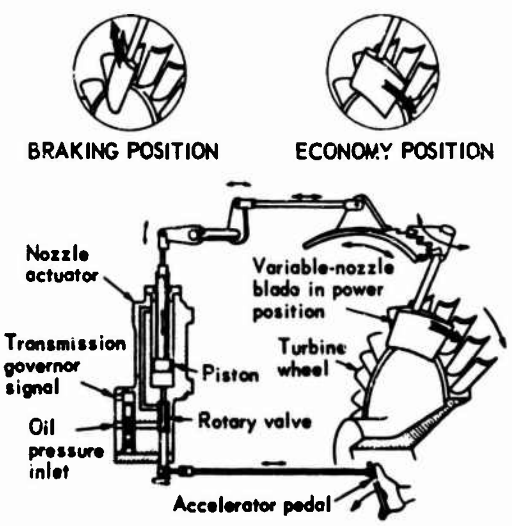


Fig. I-180—Chrysler Corporation Variable-Geometry Power-Turbine Nozzle System

The US Army AGT-1500 gas-turbine engine incorporates variable inlet guide vanes to the compressor. These vanes are remotely controlled to provide better part-load and transient performance. The use of reversing variable-geometry nozzles at the power turbine section has been proposed by several companies. This system would eliminate the use of reverse gearing in the transmission system and appears practical for large engines. This system may not be practical for smaller engines, where the complexity of the mechanical control systems required would be greater than the reverse gearing presently envisioned.

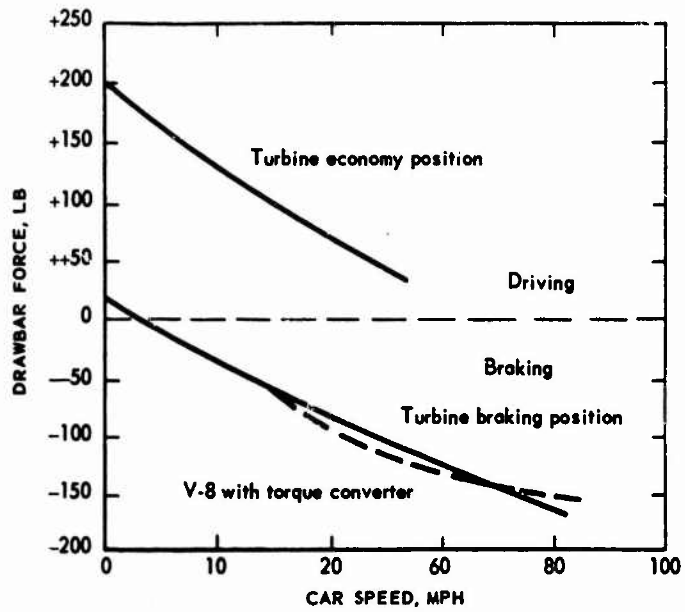


Fig. I-181—Engine Braking Characteristics of Chrysler A-831 Automotive Turbine Engine with Variable Turbine Nozzle

DIFFERENTIAL TURBINES

To approach or achieve the desired constant horsepower characteristics in a power source for automotive vehicles, many variations of the basic gas-turbine engine have been pursued. The most promising concepts appear to be the differential turbine and the split-compressor differential turbine. A schematic diagram of the basic cycle of the split-compressor differential gas turbine is shown in Fig. I-182. With this arrangement a simple gas-turbine engine consisting of one compressor (directly linked by a common shaft with the turbine) is supercharged, to an extent depending on working conditions. The addition of a device that maintains power when the power shaft is stalled permits the positive torque applied to this shaft to be increased. This is achieved by splitting the compressor in two aerodynamically coupled parts (instead of splitting the turbine). In other words a second (augmenting) compressor stage is added to the existing compressor stage, and a differential gear is introduced between the two compressor stages and the output shaft so that under normal conditions far more work is done by the compressor stage directly linked with the turbine than by the augmenting compressor stage.[3]

The basic concept of the differential gas-turbine engine, as shown in Part d of Fig. I-183, comprises a compressor and turbine mounted on separate shafts that are connected and then affixed to an output shaft through a differential gear. The planet centers are connected to the output shaft, the outer ring gear is connected to the turbine shaft, and the sun gear is connected to the compressor shaft. All are free to rotate. In the typical example shown (Part d of Fig. I-183), when the output shaft is stationary, the planet centers are stationary and the respective gears act as idlers. Thus the turbine drives the compressor at

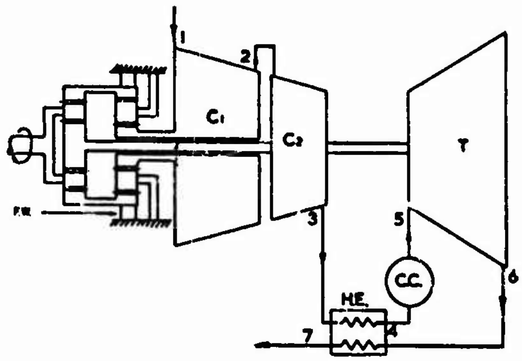
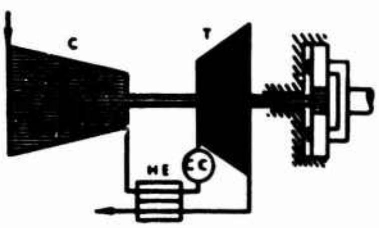
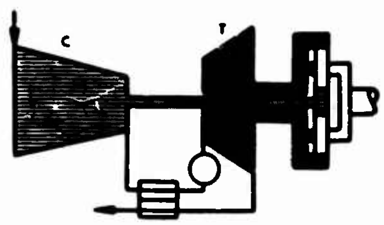


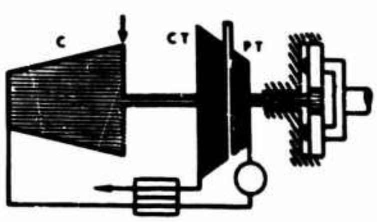
Fig. I-182—Basic Cycle of Split-Compressor Differential Gas Turbine



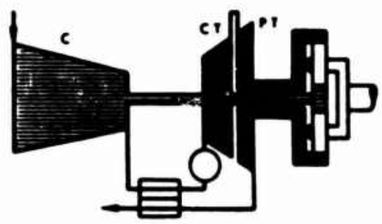
a. Simple Open-Cycle Gas Turbine



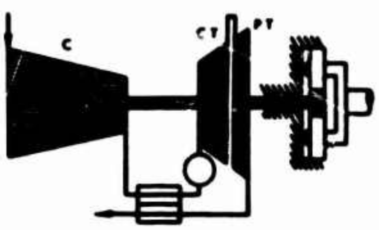
d. Simple Open-Cycle Differential Gas Turbine



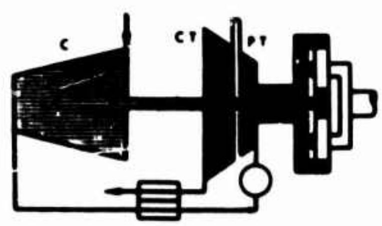
b. Open-Cycle Gas Turbine with High-Pressure Compressor Turbine and Low-Pressure Power-Turbine Output



e. Open-Cycle Gas Turbine (as in b) with Differential Gear Linking the Turbine



c. Open-Cycle Gas Turbine with Low-Pressure Compressor Turbine and High-Pressure Power-Turbine Output



f. Open-Cycle Gas Turbine (as in c) with Differential Gear Linking the Turbine

Fig. I-183—Comparison of Gas-Turbine Cycle Arrangements

C = Compressor
CT = Compressor turbine
T = Turbine
PT = Power turbine
CC = Combustion chamber
HE = Heat exchanger

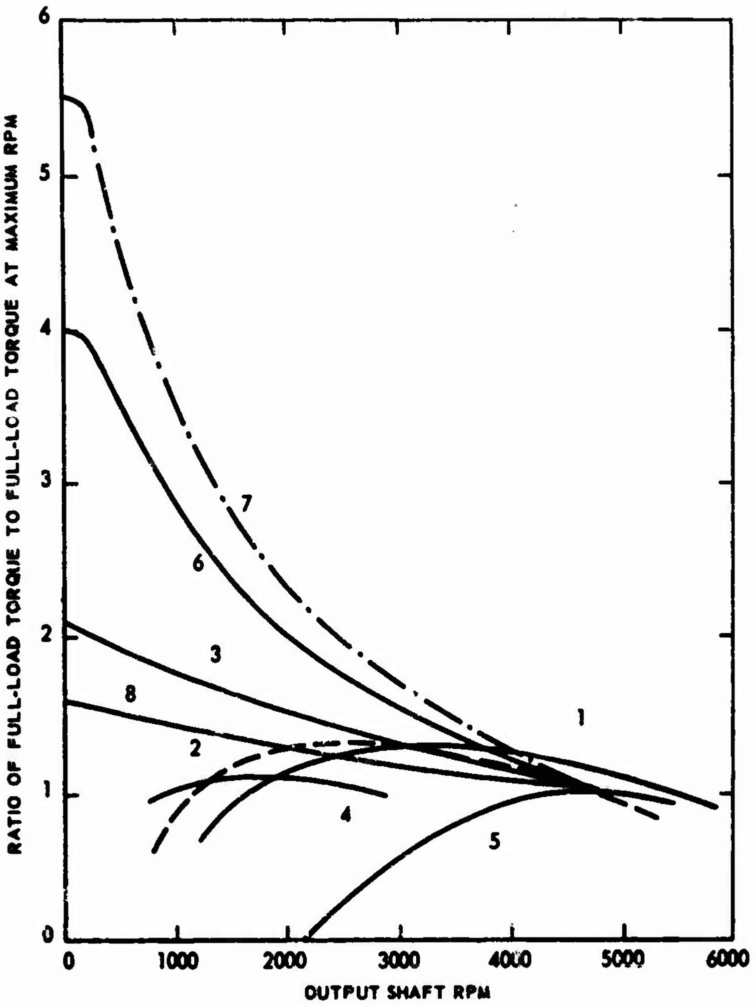


Fig. 1-184—Typical Full-Load Torque/Speed Characteristics of Various Automotive Power Sources

1. Gasoline engine
2. Wankel engine
3. Gas turbine, free-spool
4. Diesel engine
5. Gas turbine, single-shaft
6. Gas turbine, split-compressor
7. Gas turbine, split-compressor (potential development limit)
8. Gas turbine, differential

maximum speed while rotating at approximately half speed in the opposite direction. As the turbine speed increases to equal the compressor speed (still rotating in the opposite direction), the planet centers and low-speed output shaft begin to rotate in the direction of the turbine at about one-third the speed of the turbine shaft. Further, the differential gearing serves to exert half of the torque developed by the turbine on the compressor shaft, while the other half, increased by the torque multiplication of the gearing, affects the low-speed output shaft. The differential therefore ensures approximately equal distribution of turbine power between compressor and output shafts at maximum output speeds when compressor and turbine speeds are equal. As output speed reduces, a progressively larger portion of turbine power is transmitted to the compressor shaft until at zero output-shaft speed the compressor shaft receives all the turbine power.[4]

Figure I-184 illustrates typical full-load torque/speed characteristics of various automotive power sources. The curves indicate the superior torque characteristics of the split-compressor differential gas-turbine engine. The favorable torque characteristics of the differential gas-turbine engine are exceeded only by the free-spool gas turbine and the split-compressor differential gas turbine.

The advantages of the differential turbine and the split-compressor differential turbine engine are:

(a) Low SSFC at part-load and idle conditions
(b) Quick response to load changes
(c) Positive torque control
(d) Available high braking torque

Both systems, however, introduce some complexity in comparison to the conventional split-shaft turbine engine. This complexity is reduced somewhat with the elimination of some range gearing, and overall gear reduction is lowered owing to the better torque characteristics of these turbine concepts.

SINGLE-ROTOR TURBINE

The Curtiss-Wright Corporation has recently developed a gas-turbine engine in which the compressor and turbine assemblies are combined in a single rotating member. The single-rotor gas-turbine engine (see Fig. I-185) comprises a single-stage supersonic axial-flow compressor with a radial-inflow–axial-exit turbine wheel interwoven through the compressor blading and disk. Thus there is a unique combination of rotating elements within a single disk. The complete gas generator consists of stationary inlet guide vanes, the rotor,compressor exit stators and diffuser, combustion chamber, turbine nozzle assembly, and a turbine-discharge diffuser.

The experimental test engine, Model WTS-11 (shown in Fig. I-186), is designed to develop 500 to 600 shaft hp. The major components of the WTS-11 engine are shown in Figs. I-187 to I-189. The compressor, occupying the annulus near the rotor tip, is a single-stage supersonic compressor of the shock-in-rotor variety, utilizing an inlet guide vane, the rotor cascade, and an exit radial diffuser and vanes. It differs from other compressors in that the rotor cascade is completely shrouded. In other words the compressor cascade consists of holes through a solid rotor with a flow-area distribution in each passage that diverges slightly from front to rear. The compressor is designed to induce 4.75 lb of flow per second while providing a pressure ratio across rotor and inlet guide vane of 3.7 to 1 at an efficiency of 83 percent. The inlet guide vanes prerotate the inlet air opposite to the rotor direction. The trailing edges of the compressor blades are rather blunt. Because hollowed-out portions of the compressor blades serve as turbine flow passages in the rotor, this compressor is unique since it absorbs heat within its passages.

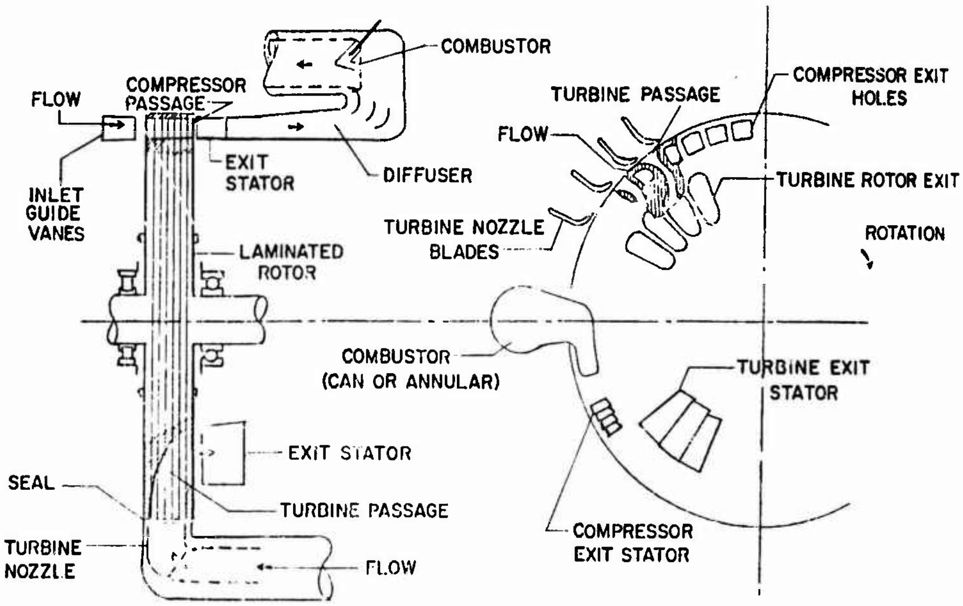


Fig. I-185—Schematic of Curtiss-Wright Single-Rotor Gas-Turbine Engine

Fig. I-186—Curtiss-Wright Model WTS-11 Single-Rotor Gas-Turbine Engine
(Test-Rig engine)

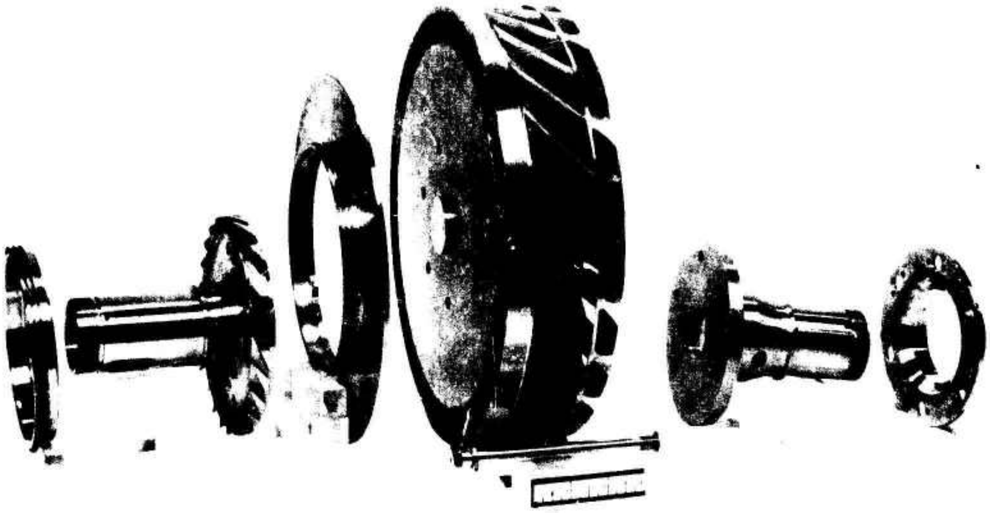


Fig. I-187—Exploded View of WTS-11 Single-Rotor Test-Engine Components

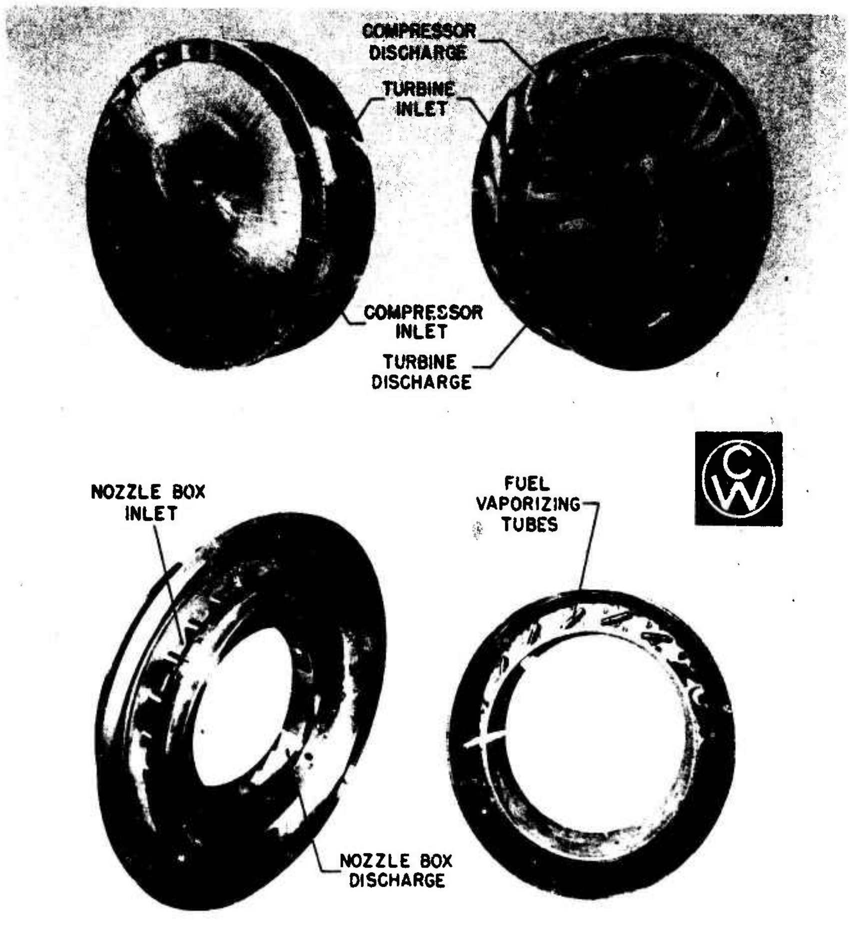


Fig. I-188—Main Components of WTS-11 Single-Rotor Turbine Engine

The Model WTS-11 turbine has a radial inflow design utilizing a sonic nozzle and a fully shrouded rotor with no exit stators. A diffusion passage to accept the exit flow is incorporated. The convection cooling provided by the compressor permits a turbine-entry design temperature of 2800°R (2340°F). The flow is essentially that induced by the compressor, less minor leakage, with the addition of combustor fuel. Although the compressor cooling permits the entry temperatures to exceed those of contemporary engines at the leading edges of the turbine rotor blades, the radially-inward portions of the turbine passages require cooling because turbine exit temperatures are still quite high.

Fig. I-189—View of Turbine-Discharge Side of WTS-11 Engine Rotor

To provide this cooling it is necessary to design cooling flow passages through the rotor to permit convection cooling of the turbine walls. This is accomplished by a rotor-pumped air supply that flows through the rotor and is exhausted into the turbine exit flow. A high degree of turbine cooling is achieved since all the gas generator mass flow is available for convection cooling of the turbine portion of the rotor. The operating cycle should permit turbine temperatures far in excess of those of contemporary engines.[5]

Testing of components and of the complete Model WTS-11 engine is currently being conducted. The developers of the single-rotor turbine concept believe a good understanding of the operating characteristics of the compressor, turbine, and combustion chamber exists. Operation, at present, is at lower than design turbine-inlet temperature. The developers also estimate that with adequate support funds, the single-rotor turbine engine could eventually be developed to deliver 150 to 160 hp/lb air/sec and operate at turbine inlet temperatures of 2400 to 2500°F.

Because of the close proximity of the compressor and turbine, and the complex flow paths of air and gas, attainment of adequate heat-transfer control will require a large amount of development work. Also because of the airflow paths required by the configuration of this engine, thick, sturdy, nonoptimum airfoil must be used, and consequently component efficiencies will be lower than those of conventional turbine engines. Low SSFC will be difficult to attain, even with regeneration. The major advantages of the single-rotor engine will be high power-to-weight and power-to-size ratios and low production cost. The characteristics of this engine make it ideal as a power-boost or topping engine where extremely high power-to-weight ratios are required, as in large combat assault vehicles.

SIAMESED-TURBINE CONCEPT

A recent unique application of gas-turbine engine principles to an integrated engine-transmission system is shown in Fig. I-190. This system, proposed by the Southwest Research Institute, is referred to as the "siamesed turbine concept." In this system the compressor is split into two separate components: one component to supply air for the gas-generator section, and another to supply air for the power section of the engine. The two separate systems are selectively coupled by an arrangement of hydromechanical and aerodynamic links. The concept might also be thought of as a pneumatic transmission system, since the concept is analogous to the system used in the land train, with electricity being replaced by air. A central turbine drives an air compressor (rather than a generator) and air (rather than electricity) is distributed to individual power turbines (rather than electric motors) at each driving wheel.

Two separate systems can be selectively coupled through a fluid coupling or hydromechanical clutch. The central system consists of a single-spool recuperated (or regenerated) gas-turbine engine whose output shaft is joined to the input side of the fluid coupling. The power system consists of a single compressor (driven by the central system through the fluid coupling) that supplies air to individual burner-turbine components at each drive sprocket or drive wheel of the vehicle. The entire power system can be completely inoperative while the central system is operating at any desired speed.[6] Reverse vehicle operation is accomplished by reversing the power turbines through the use of reversing nozzle rings or reversible variable-geometry nozzles.

The advantages claimed for this power system are:

(a) Reduced low-power and idle fuel consumption

(b) Rapid engine response

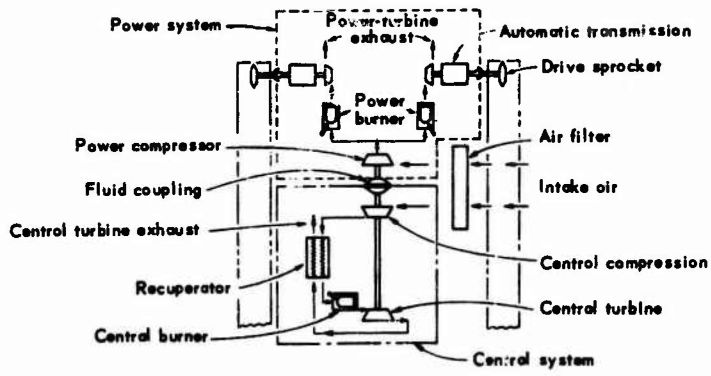


Fig. I-190—Schematic Diagram of a Possible Siamesed Turbine

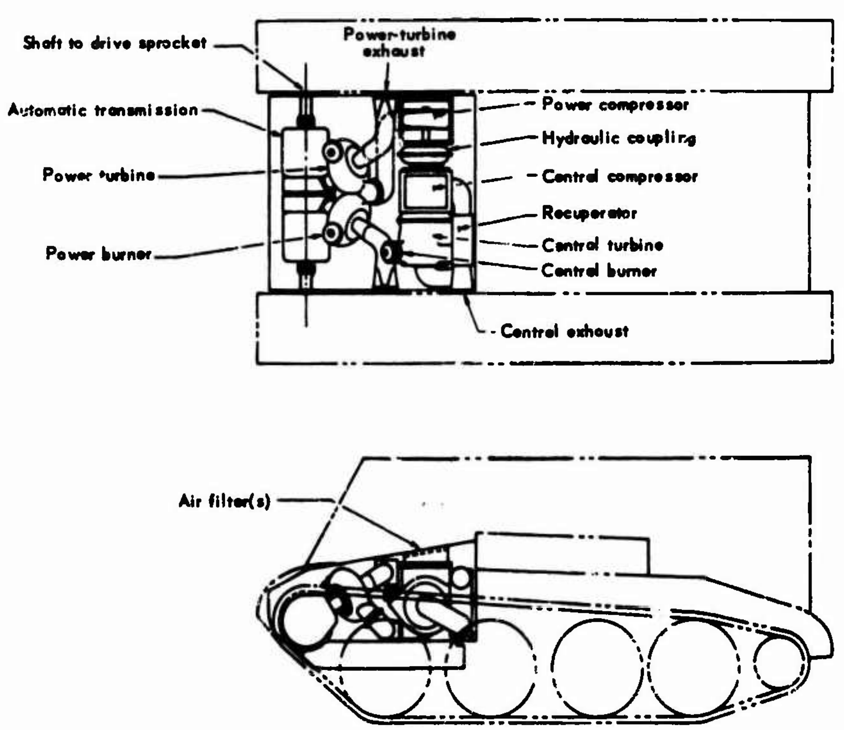


Fig. I-191—Possible Installation Arrangement of Siamesed-Turbine System in Light Full-Tracked Vehicle

(c) Engine reversing capability
(d) Availability of individual power turbine for steering augmentation
(e) Available high braking torque
(f) Ease of maintenance and replacement

A possible installation of a siamesed-turbine system in a light full-tracked vehicle is illustrated in Fig. I-191.

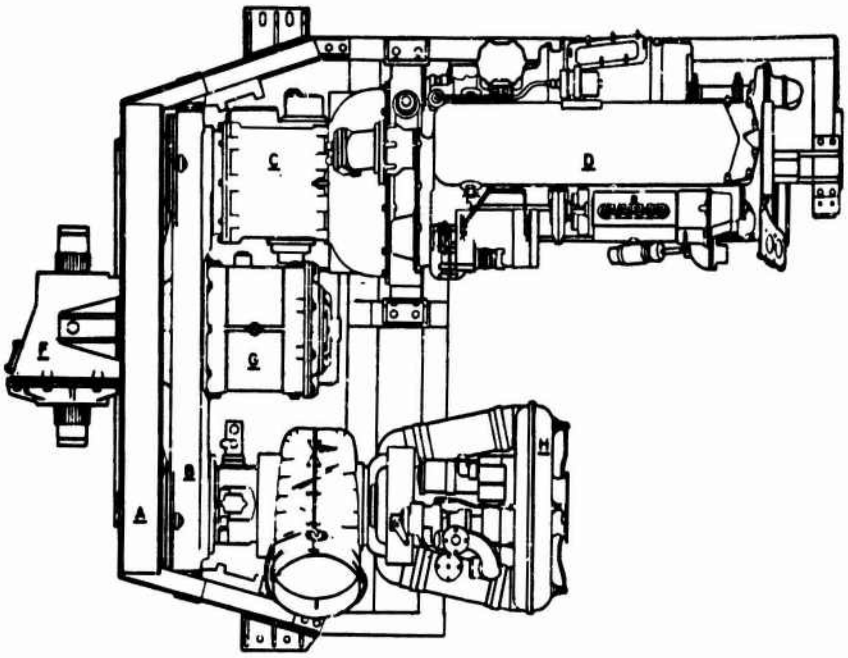
COMPOUND POWER-BOOST SYSTEM

The compound power-boost system may well be the most feasible approach to meet increased power requirements within a limited space-weight package, such as would be required for future combat assault vehicles. The compound power-boost system is defined as any system in which the power outputs from two or more separate and independent power sources are coupled in a single output. The power sources may be a compression-ignition engine coupled with an unregenerative gas-turbine engine, or a regenerative gas-turbine engine coupled with an unregenerative high-output gas turbine.

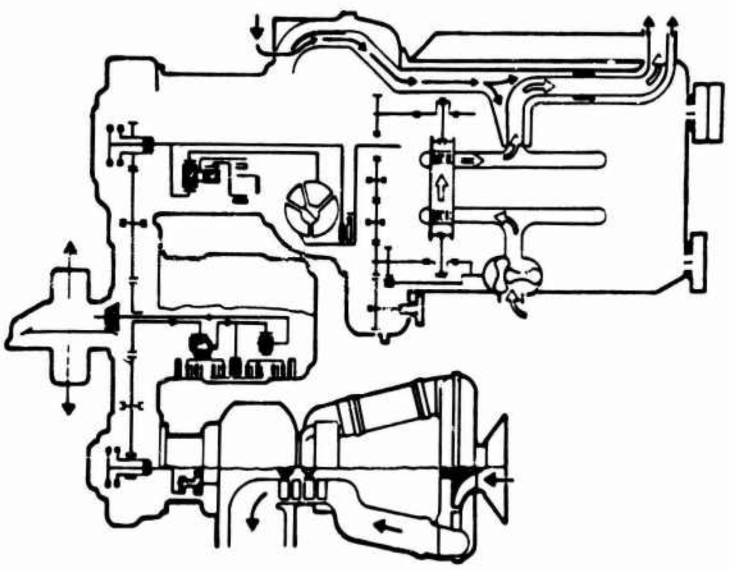
A feasibility study program conducted in 1960 concluded that a high-output compression-ignition engine teamed with a simple-cycle free-shaft gas-turbine engine was the optimum combination. The objectives of the project were to combine the fuel economy of the diesel engine with the advantageous weight, size, torque, and cold-starting characteristics of the gas-turbine engine. In most military vehicular applications the operating duty-cycle is such that the engine is operated at a relatively low load factor most of the time. Generally engines operate more efficiently at full-load output than at part-load output. By utilizing two engines, as in the turbine power boost, the main (diesel) engine can be operated at a more efficient higher load factor during normal vehicle operation. The turbine can be used as a high-power topping engine in combination with the diesel engine when vehicle operation demands peak power.

A detailed analytical study was made to compare the performance characteristics of a compound system comprising a compression-ignition engine coupled with a free-wheeling gas-turbine engine with those of a large high-output compression-ignition engine. Studies were also made to establish optimum drive-train design and methods of obtaining rapid response from the turbine engine. (The slow response of the gas-turbine engine has been one of its drawbacks as a vehicular propulsion device. However, recent developments have minimized this trait considerably.) Achieving complete control of both engines during tandem operation has presented some problems, and work is progressing in this area.

Recent studies indicated that the most efficient system would be a combination of two types of engines rated at nearly equal horsepower. The exceptional reliability of this system is made possible by the fact that the turbine, which operates a small percentage of the time (approximately 10 percent), is in effect a spare engine with dependable cold-starting characteristics. Study results show that for a battlefield day, the turbine boost engine can be operated 10 percent of the time in conjunction with the base diesel engine without exceeding the fuel consumption of a single large high-output compression-ignition engine.



a. Main Component Arrangement



b. Design Principle of Dual Power Plant

Fig. I-192—Volvo Compound Diesel-Turbine Power System

A twin-turbine installation was made in a T42 test-bed tracked vehicle. The engines were of the unregenerative, free-spool type, rated at approximately 300 hp each. After some testing, one turbine engine was removed and replaced by a commercial compression-ignition engine of similar power. The primary purpose of this installation was to study the problems of combining two distinctly different power sources with respect to power transmission and control of the engines.

Several combinations of engines have been compared with respect to performance, fuel economy, multifuel capability, cold-start capability, range, dependability, durability, compactness, weight, simplicity, producibility, and cost. The studies show that the power-boost system is advantageous in power ranges above 1000 hp. A successful turbine power-boost system has been incorporated in a medium-class assault-gun tank, now in limited production in Sweden. This power package, shown in Fig. I-192, couples a 240-hp compression-ignition multifuel engine with a 330-hp simple-cycle free-shaft gas-turbine engine. This compound system appears to have all the advantages listed above. Figure I-193 illustrates the compound diesel-turbine power system installed in the Swedish S-tank.

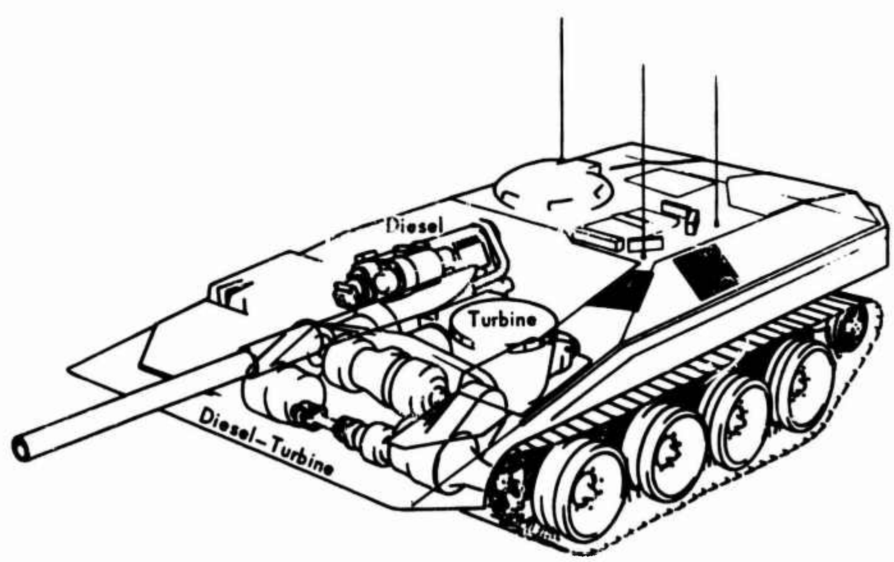


Fig. I-193—Compound Diesel-Turbine Power System Installed in Swedish S-Tank

The compound power-boost system, using advanced-technology compression-ignition and gas-turbine engines, could result in a very lightweight and compact power system. It is estimated that the power-boost system in the power range of 1500 to 2500 hp would have a specific weight of approximately 1 lb/hp and a specific power output greater than 50 hp/ft^3. However, the compound system would be much more costly than either a single large compression-ignition or turbine engine.

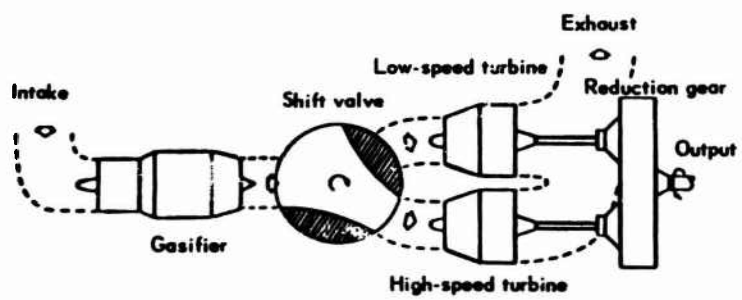


Fig. I-194—Single Gasifier Driving Twin-Turbine Transmission

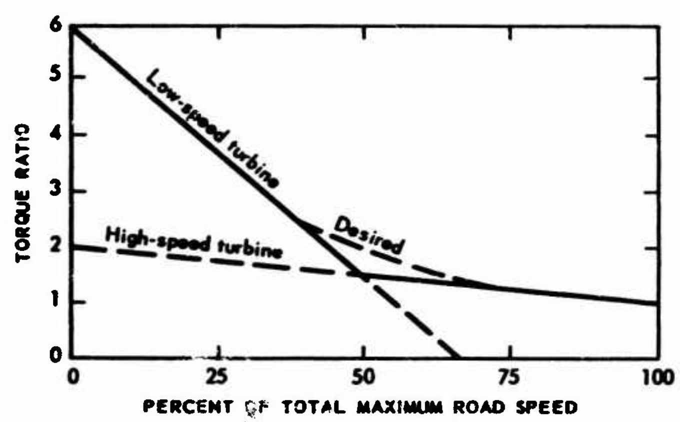


Fig. I-195—Possible Torque-Speed Characteristics of Twin-Turbine Power System

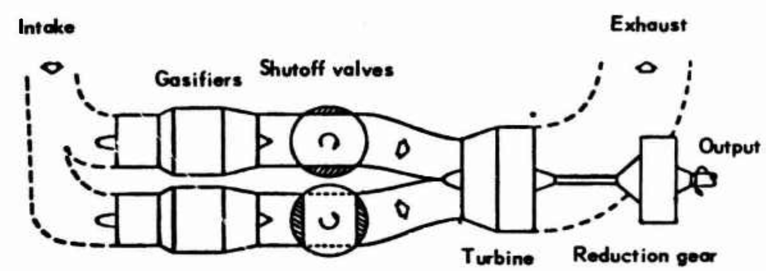


Fig. I-196—Twin Gasifiers Driving a Split-Entry Turbine

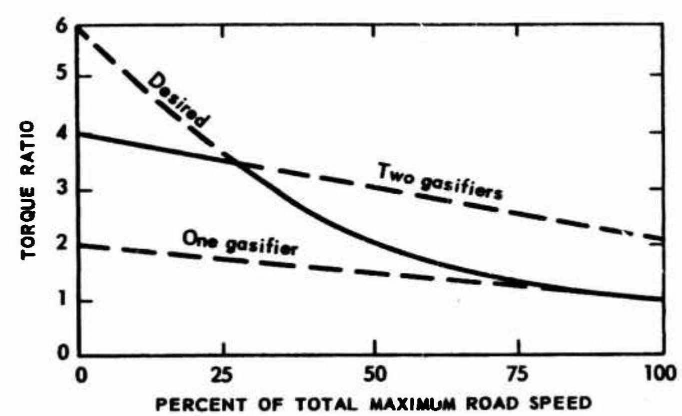


Fig. I-197—Possible Torque-Speed Characteristics of Twin-Gasifier Power System

TWIN-TURBINE AND TWIN-GASIFIER ENGINES

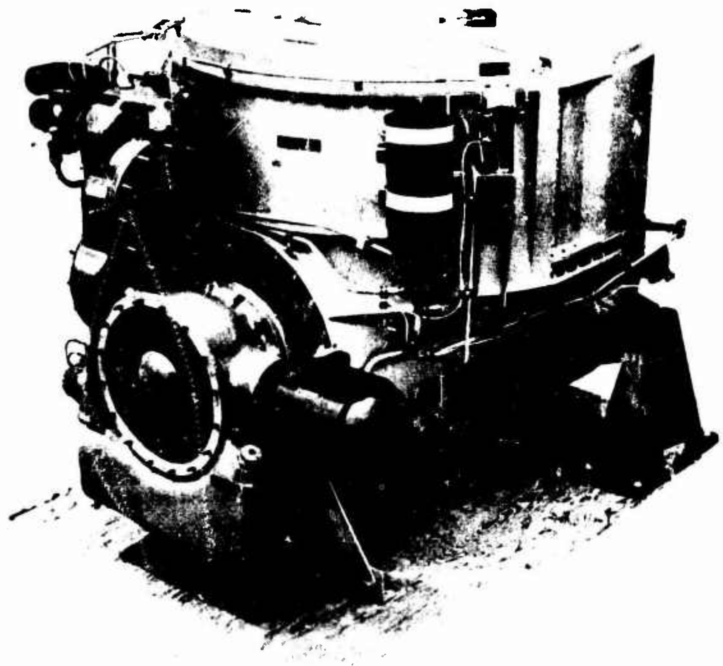
A unique application of the gas-turbine engine to achieve or approach desired constant-horsepower characteristics for the powering of automotive vehicles is shown in Fig. I-194. This system utilizes a gas-turbine engine as a gasifier unit to supply a low-speed turbine and a high-speed turbine. Gas flow is controlled by a large valve. A possible torque-speed output plot of this system is shown in Fig. I-195. Figure I-196 illustrates a system that utilizes twin gasifiers to feed a single turbine. Each gasifier unit is controlled by a valve. A possible torque-speed plot of this system used with either or both of the gasifiers operating is shown in Fig. I-197. The unique feature of these systems is the elimination of the conventional torque converter and range-gear transmission. Only a reduction-gear unit is required. Torque conversion is accomplished aerodynamically. The aerodynamic torque converter (and the absence of the range-gear transmission) results in a much lighter power system. The volume of either system would be comparable to that of a conventional power system. The twin-gasifier system (Fig. I-196) would be much more costly than a single-engine unit, but its reliability would be much greater.

A similar system was developed by the Allison Division of GMC in 1960, under contract to ATAC. This concept, called the "Gasamatic" system, utilized three gas-turbine engines as a gasifier supplying twin free-wheeling power turbines. It was anticipated at the time that this system would eliminate the conventional torque converter and still maintain the necessary torque multiplication. Tests of the prototype unit indicated that the stall torque and torque coverage ratio were much less than first estimated, and the program was terminated.

MILITARY-SPONSORED ENGINES

There are no fully developed gas-turbine engines, nor any sufficiently far along in development to be considered, available for use in an automotive-type military vehicle today. There are several commercially developed unregenerated engines in power ranges below 400 hp that have seen some application with a fair degree of success. The experimental regenerative gas-turbine engines being developed by the automobile manufacturers have not been sufficiently successful to warrant application to military vehicles. These engines have reportedly demonstrated good performance in passenger cars and trucks but are far from being ready for production.

The Army-Navy-sponsored development program for the 600-hp regenerative turbine engine has resulted in units undoubtedly better suited for tactical vehicle application, but they too require much more development. Under this contract, the Solar Aircraft Division of International Harvester Corporation developed a free-spool simple regenerative-cycle engine with variable-geometry turbine nozzles and a rotary heat exchanger. This engine, designated the Solar Model T-600, is shown in Fig. I-198. Development of the Solar engine was terminated before an operable engine could be assembled. However, design concentration on regenerator development continued for a period of time. The Solar Aircraft Division is continuing development of the T-600 engine without Government sponsorship. The application is intended for large transcontinental commercial trucks.



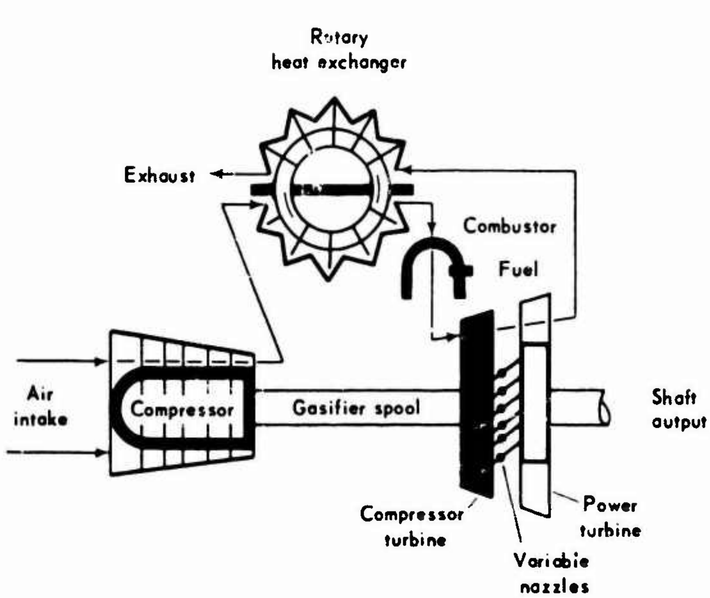


Fig. I-198—Solar Model T-600 600-hp Gas-Turbine Engine
Simple regenerative cycle.

The Ford Motor Company has developed a complex, or turbocharged-cycle, free-spool engine that incorporates intercooling, reheat, and recuperation stages. This engine, designated the Ford Model 705 (see Fig. I-199), is based on an earlier Ford engine, Model 704, that developed 300 hp.

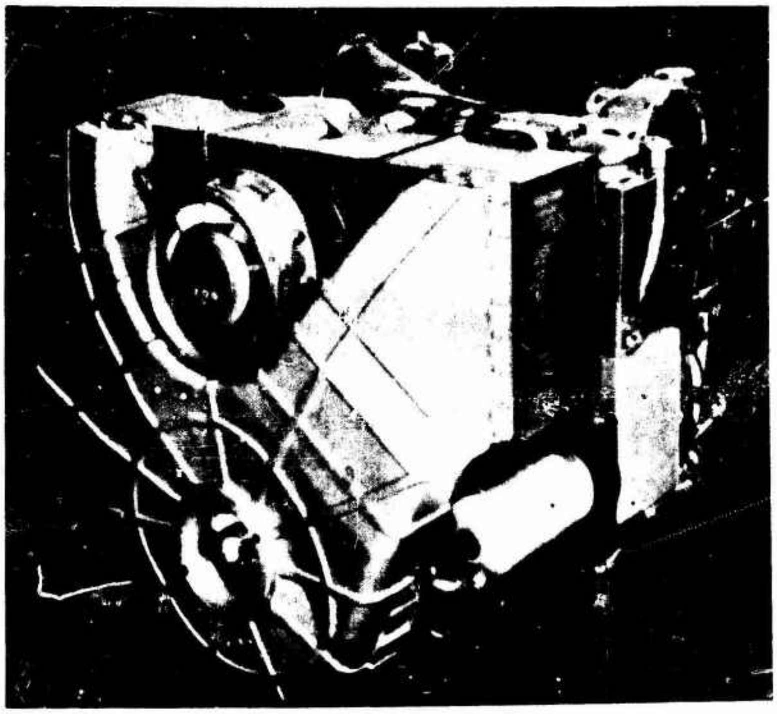
The Ford Motor Company continued work, with corporate funding, by building and testing several of these engines in large trucks. Development work on the Model 705 engine has since been discontinued because Ford believed that the advent of ceramic heat-exchanger materials made the complex-cycle engine concept obsolete.

Orenda Engines, Ltd., of Canada, has developed a simple-cycle free-spool recuperated engine with variable-geometry turbine nozzles and a stationary heat exchanger. This engine, shown in Figs. I-142 and I-200, is designated the Orenda Model OT-4. Several of these units were completed for test. Two engines were installed in M48 full-tracked test beds and are currently undergoing test. Many problem areas have been encountered, but overall the OT-4 engine appears to be performing satisfactorily. Full development of the 600-hp turbine engines has been terminated owing to the increased power requirements of the main battle tank.

A comparison of the characteristics of the Army-Navy-sponsored 600-hp gas-turbine engines is shown in Table I-20. Figure I-201 illustrates a comparison of estimated fuel consumption. The Ford turbocharged-cycle engine has much better part-load fuel economy than the simple-cycle Solar and Orenda engines. However, the Solar and Orenda engines have better full-load fuel economy.

The passenger-car gas-turbine engine developed by the Chrysler Corporation is a simple-cycle regenerative engine. Figure I-202 illustrates the road-load fuel economy of this engine installed in a passenger car weighing approximately 4200 to 4400 lb. The fuel curve indicates a fuel economy comparable with that of a conventional reciprocating-piston spark-ignition engine. The Chrysler A-831 engine operates at a turbine inlet temperature of 1700°F and develops 130 shp. A later version of this engine operates at a turbine-inlet temperature of approximately 1760°F and develops 160 shp. The engine weighs approximately 410 lb and has a volume of approximately 17 ft^3. The specific power output of the 160-shp unit (which develops an estimated 180 gross hp) is 10 to 11 hp/ft^3, and its specific weight is approximately 2.3 lb/hp.

The Ford Motor Company is currently developing two new low-pressure simple-cycle split-shaft regenerative gas-turbine engines. These engines incorporate twin ceramic rotary heat exchangers. The smaller engine, designated the Model 706, develops approximately 200 hp. This engine is an experimental passenger-car unit. The large engine, the Model 707, is of the same basic design and develops approximately 375 hp. The engines operate at turbine-inlet temperatures of approximately 1750°F, uncooled. The Ford Motor Company stated that the Model 706 engine has bettered the fuel economy of a comparable conventional spark-ignition engine. The goal for the large engine is to better the fuel economy of a comparable diesel engine by approximately 8 percent. The engines are simple in design and should lend themselves to low-cost production, i.e., the small engine being comparable in cost to the conventional piston engine, and the large engine being comparable in cost to the conventional diesel engine, if either is produced in comparable quantities. Both engines appear to be bulky and heavy. It is estimated that the specific output



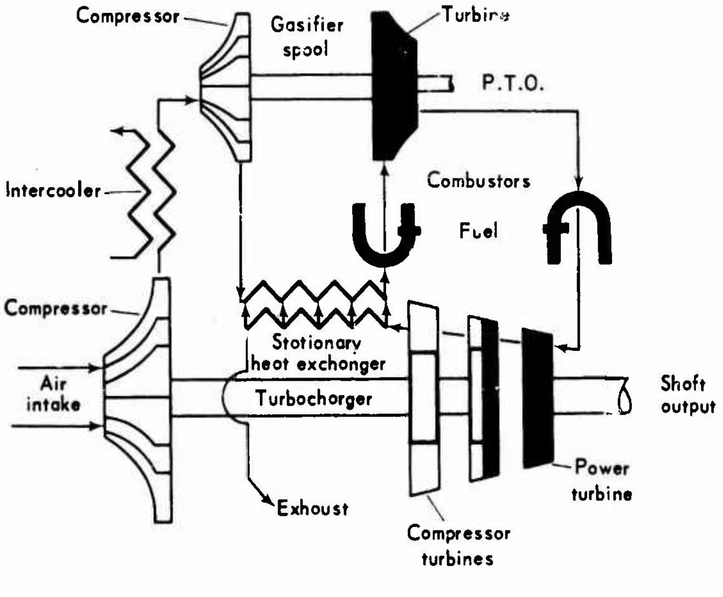
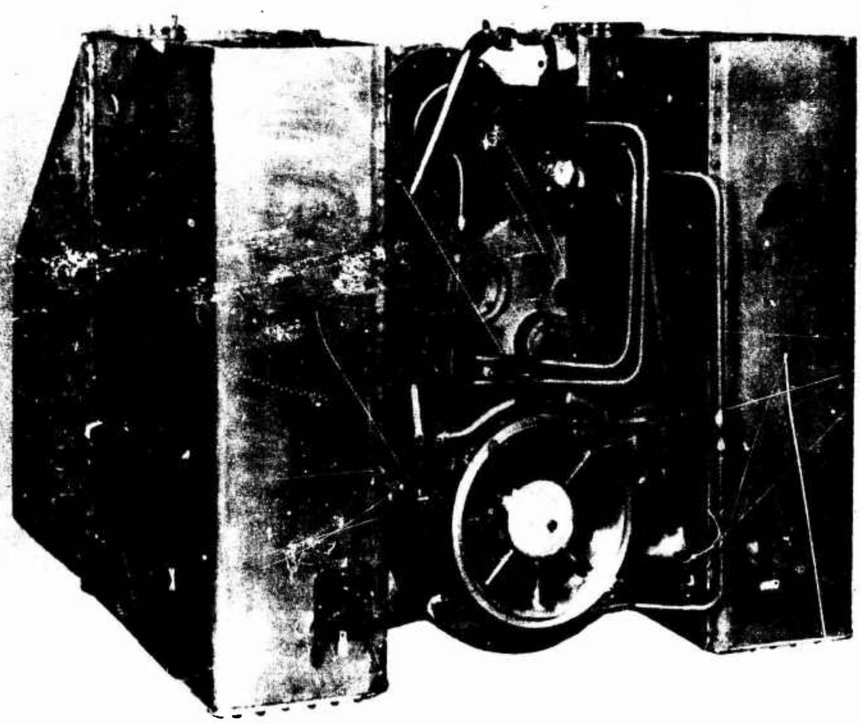


Fig. I-199—Ford Model 705 600-hp Gas-Turbine Engine
Turbocharged recuperative cycle.



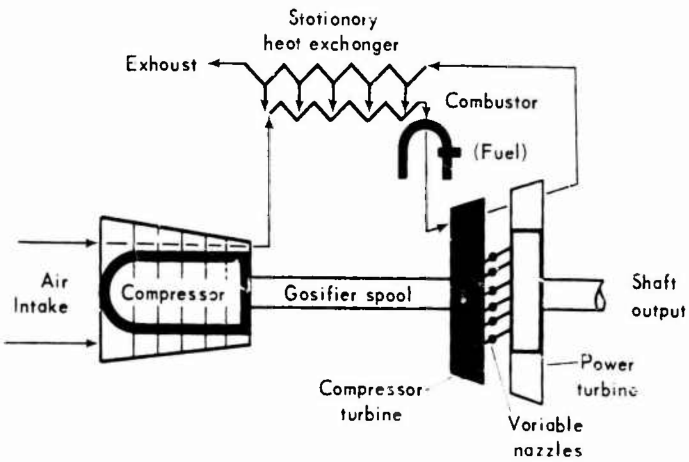


Fig. I-200—Orenda Model OT-4 600-hp Gas-Turbine Engine
Simple recuperative cycle.

TABLE 1-20

Comparison of Characteristics of Army-Navy-Sponsored 600-hp Gas-Turbine Engines[a]

Item	Ford 705	Solar T-600	Orenda OT-4
Air flow, lb/sec	4.4	8.5	6.5
Pressure ratio	16	3.8	4.0
Compressor stages	2 centrifugal	6 axial	6 axial
Efficiency, %	80	87	86
Outside air to intercooler, lb/sec	4.4	—	—
Heat exchanger	Recuperator	Regenerator	Recuperator
Effectiveness, %	80	89	82
Leakage loss, %	—	5.5	—
Burner discharge temperature, °F	1,750	1,600	1,735
Gas-producer turbines			
Stages	1 radial, 2 axial	1 axial	1 axial
Efficiency, %	87–88	89	89
Gas producer, max. rpm	75,000	21,500	26,800
Power-turbine stages	1 axial	1 axial	1 axial
Efficiency, %	88	89	89
Variable power-turbine nozzles	No	Yes	Yes
Sum of cycle pressure losses, %	23	16	16
Engine volume, ft^3	48	60	58
Engine weight, lb	1,200	1,500	1,450
Specific output, hp/ft^3	12.5	10.0	10.3
Specific weight, lb/hp	2.0	2.5	2.4

[a]All efficiencies are total to total.

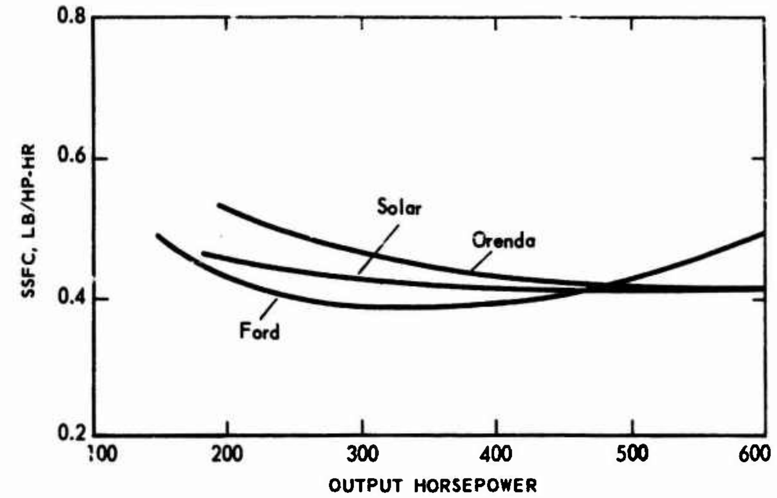


Fig. I-201—Estimated SSFC of Army-Navy-Sponsored 600-hp Gas-Turbine Engines

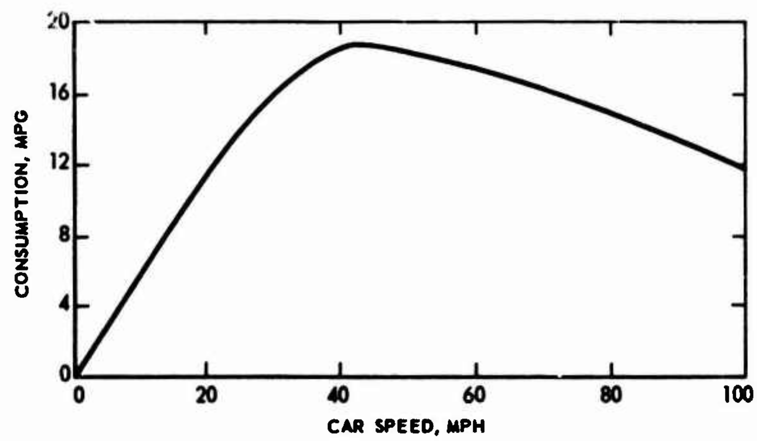


Fig. I-202—Road-Load Fuel Consumption of Chrysler Passenger-Car Powered by Model A-831 Gas-Turbine Engine of the 706 engine is $8\frac{1}{2}$ to 9 hp/ft^3, and that of the larger 707 engine is 9 to 10 hp/ft^3. The specific weight of these engines is approximately 4 to $4\frac{1}{2}$ lb/hp. This specific weight figure could be reduced somewhat if the main engine castings, now cast iron, were made of aluminum.

The US Army development gas-turbine tank engine, under development by the Lycoming Division of Avco Corporation, is an integrated in-line free-turbine-shaft engine utilizing a two-spool axial compressor, a two-stage axial gas-producer turbine, and a two-stage axial power turbine. The engine incorporates a counterflow wave-plate regenerator. The engine is designated as the Army AGT-1500 and by Lycoming Division as PLT-25. The engine is illustrated in Figs. I-203 and I-204, and Fig. I-205 shows a cross section of the same engine. (A schematic cross section is shown in Fig. I-152.) The principal characteristics of the AGT-1500 turbine are shown in Table I-21, and the cycle performance characteristics are shown in Fig. I-206. The graph indicates that the AGT-1500 engine operates at a maximum turbine-inlet temperature of 2180°F, a pressure ratio of 10.5 to 1, and delivers 172 shp/lb air/sec. This performance is about 75 to 100 percent better than that of present automotive engines.

TABLE I-21

Characteristics of Army AGT-1500 (Lycoming PLT-25), 1500-hp Gas-Turbine Tank Engine

Item	Value or description
Horsepower	1500
Output-shaft speed, rpm	3000
Pressure ratio	10.5
Air flow, lb/sec	8.84
Compressor	2-spool axial with variable-geometry inlet guide vanes
Gas-producer turbine	2-stage axial
Power turbine	2-stage axial, free-spool, with variable-geometry stators
Heat exchanger	Stationary
SSFC, lb/bhp-hr	
100% power	0.38
80% power	0.37
40% power	0.39
Idle (35 hp)	1.14
Overhaul interval, hr	1000
Weight, lb	1600
Dimensions, in.	
L × W × H	59.5 × 39.5 × 28.0
Volume, ft^3	38.3
Specific weight, lb/hp	1.07
Specific output, hp/lb air/sec	172
Specific output, hp/ft^3	39

The performance characteristics of the AGT-1500 gas-turbine engine are shown in Figs. I-207 to I-209. The fuel-consumption rate of 0.355 lb/hp-hr is at optimum output-speed.[7] However, it is impractical to operate a vehicle at

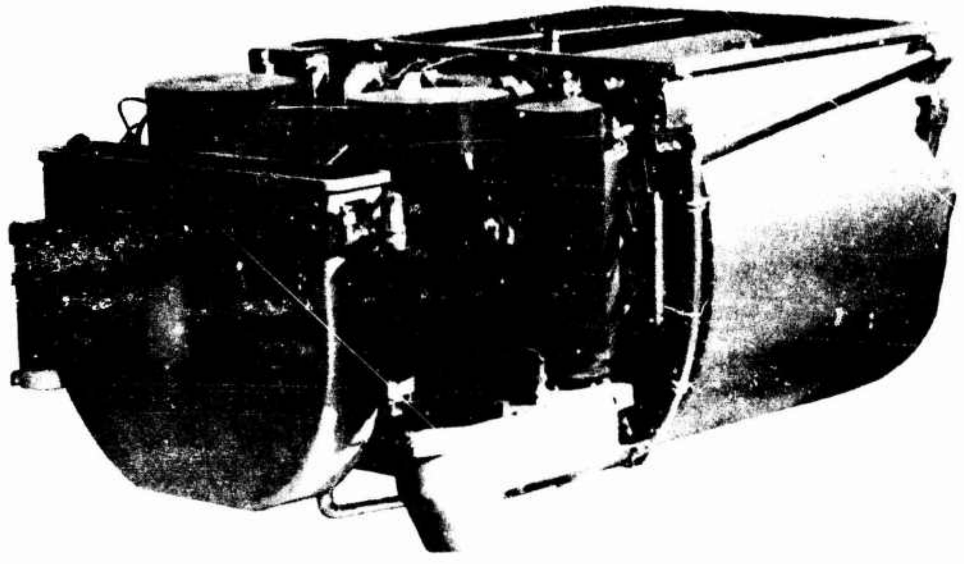


Fig. I-203—Right-Side View of Army AGT-1500, 1500-hp Recuperated Gas-Turbine Tank Engine

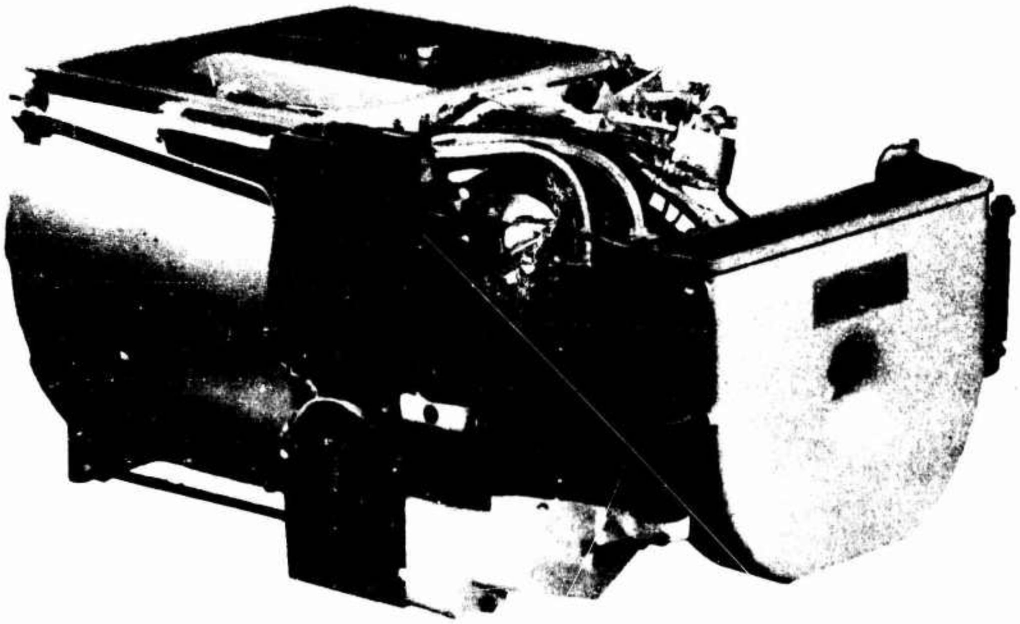


Fig. I-204—Left-Side View of Army AGT-1500, 1500-hp Recuperated Gas-Turbine Tank Engine

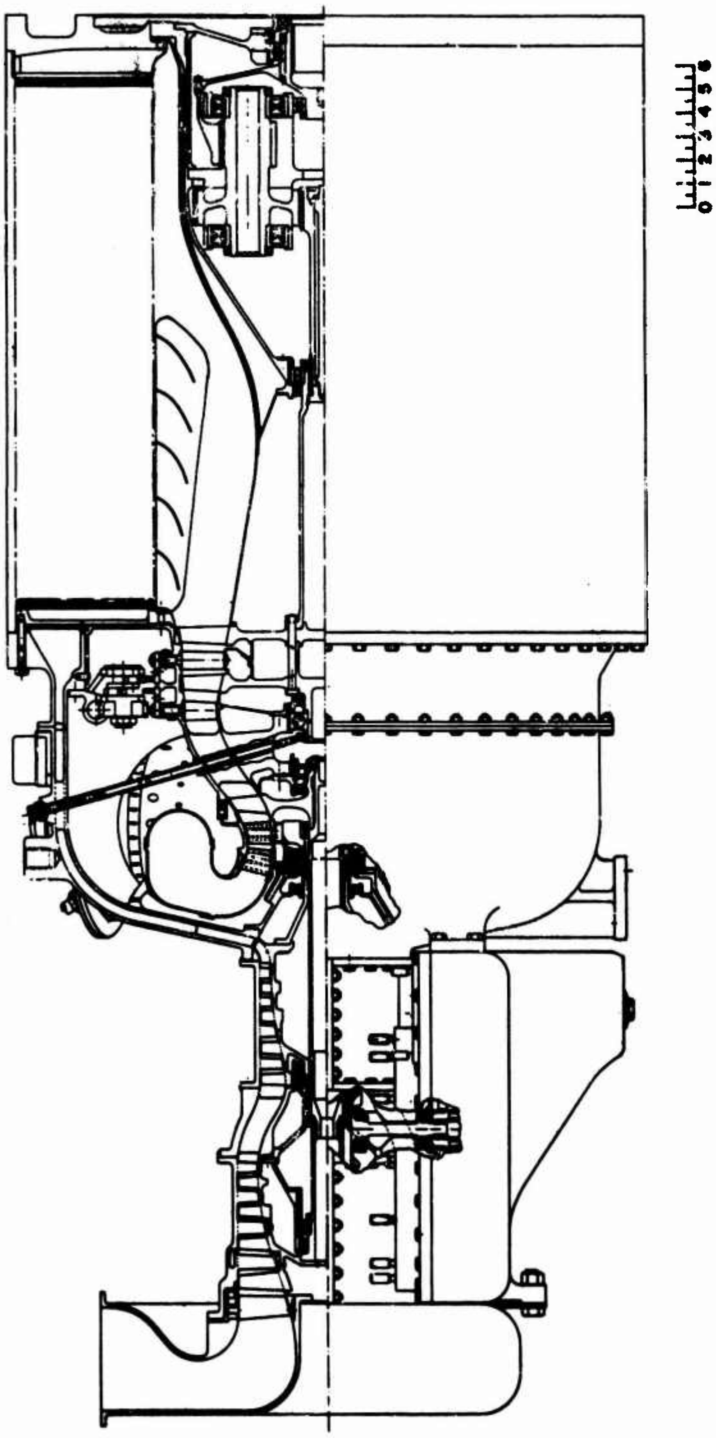


Fig. I-205—Cross Section of AGT-1500 Gas-Turbine Engine

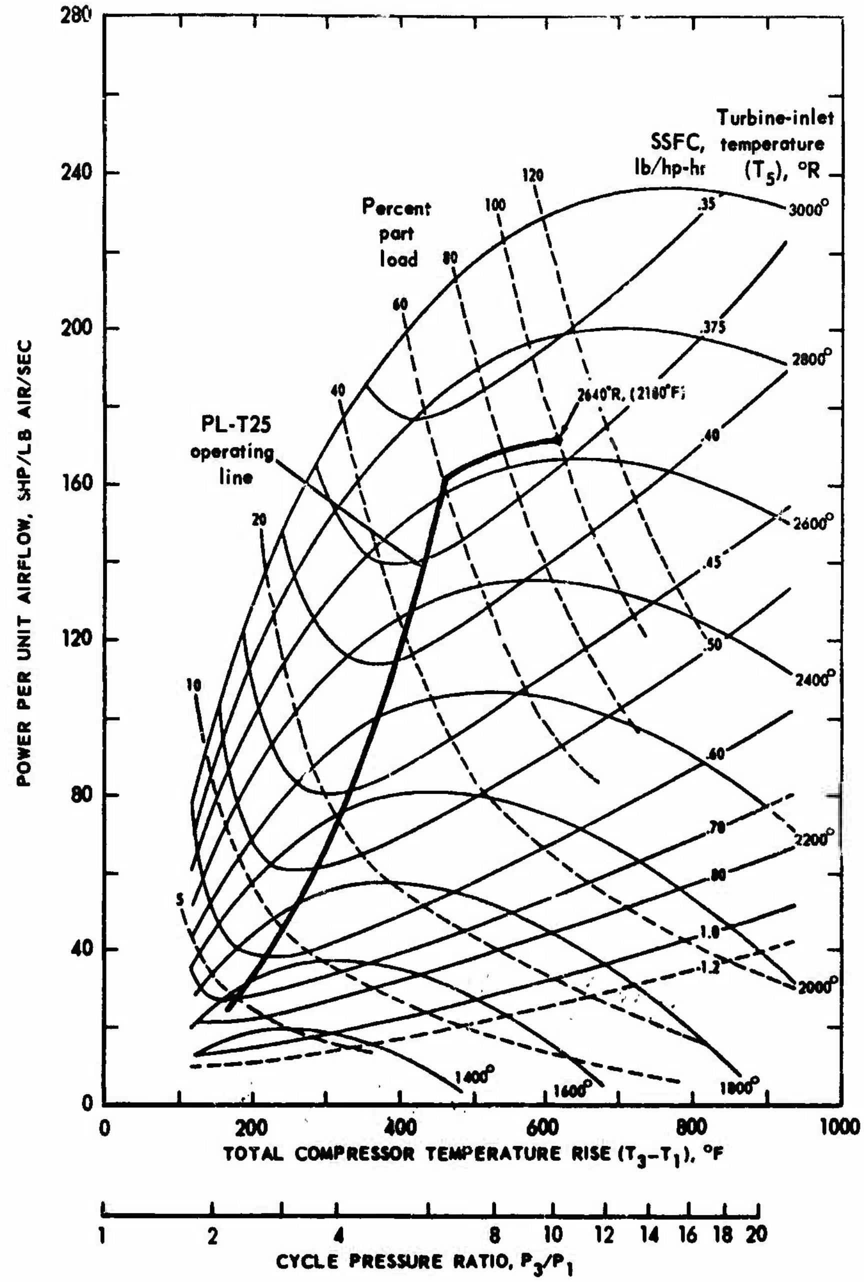


Fig. I-206—Cycle Performance for AGT-1500 Gas-Turbine Engine

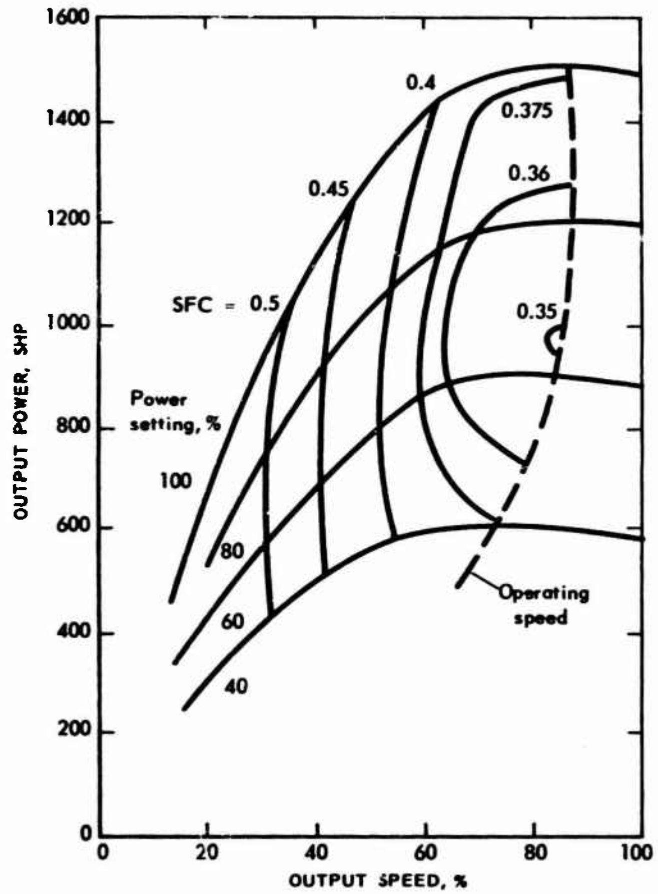


Fig. I-207—Performance Characteristics of AGT-1500 Gas-Turbine Engine: Output Power as a Function of Output Speed (Standard Conditions)

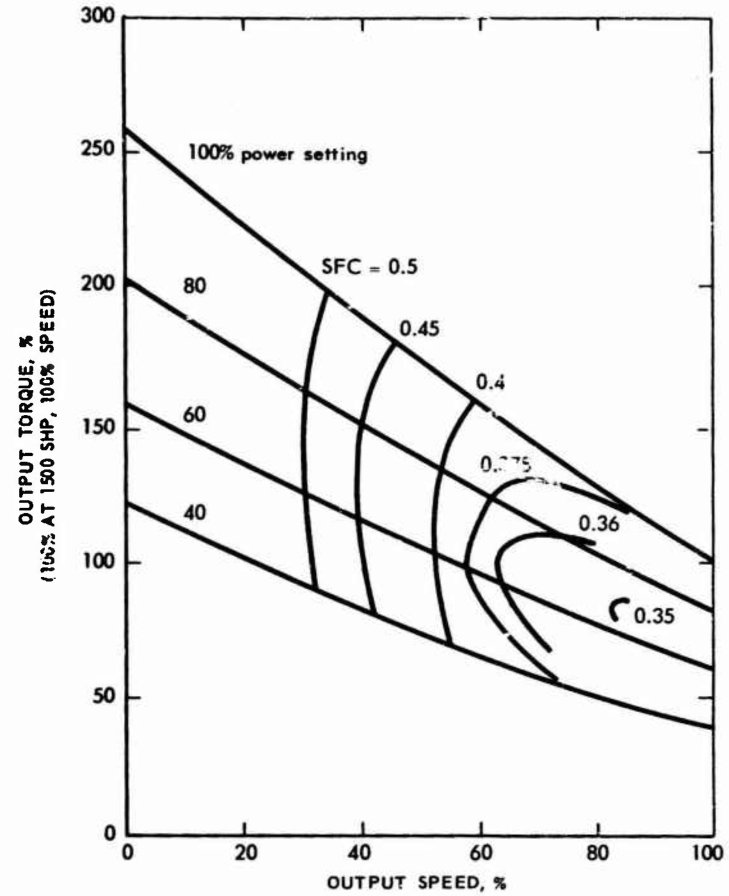


Fig. I-208—Performance Characteristics of AGT-1500 Gas-Turbine Engine: Output Torque as a Function of Output Speed (Standard Conditions) these optimum speeds to achieve the low fuel rate. The islands in the fuel map (shown in Figs. I-207 and I-208) that yield the low fuel rates are very small. A practical minimum fuel rate at part-load operation is in the region of 0.37 to 0.38 lb/hp-hr. This fuel rate is equal to that of the best compression-ignition engines.

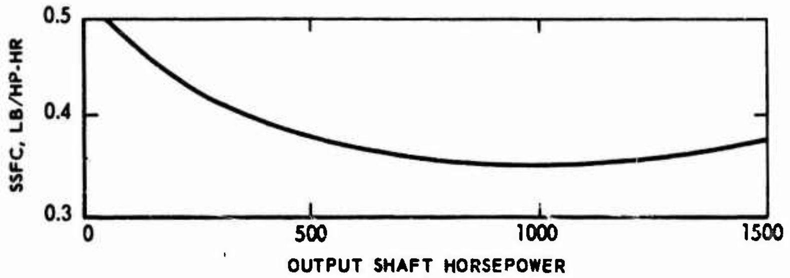


Fig. I-209—Performance Characteristics of AGT-1500 Gas-Turbine Engine: SSFC at Optimum Output Speed (Standard Conditions)

The specific weight of the AGT-1500 turbine engine is 1.07 lb/hp. This can be reduced somewhat as the turbine-inlet temperature increases to 2300 to 2400°F with advanced cooling techniques. The specific power output of this unit is 172 hp/lb air/sec and 39 hp/ft^3. These outputs are excellent and represent a 75 to 100 percent increase over present engines.

As advanced technology in component design is applied to gas-turbine engines, specific power outputs will greatly increase and fuel consumption will be further reduced.

DISCUSSION

The advantages offered by a gas-turbine engine for automotive vehicular application are:

(a) Excellent output-torque characteristics
(b) Internal air cooling, thus no requirement for antifreeze fluids
(c) Lightweight and compact
(d) Nontoxic exhaust
(e) Minimal burning, dilution, or contamination of oil
(f) Excellent low-temperature starting characteristics
(g) Multifuel capability
(h) Reduced logistics demands

The output-torque characteristics of the split-shaft engine are ideal for automotive applications. The two-shaft gas-turbine engine has characteristics similar to those of a reciprocating engine with a hydraulic torque converter without the disadvantages of extra weight and bulk of the housing and fluid demanded by the torque-converter element.

The gas-turbine engine has internal air cooling and does not require a separate cooling system with radiators, fans, air ducting, and other associated components (hoses, thermostats, belts, pulleys, shafts, etc.) as do both the air-cooled and liquid-cooled piston engines. Maintenance is thereby simplified, since these components are usually high-maintenance items. Also the gas-turbine engine does not require the use of antifreeze fluids, a feature that further reduces the logistics burden.

The gas-turbine engine is lighter and smaller than a comparable compression-ignition engine. Although the gas-turbine engine does require use of bulky air-filtration units and inlet and exhaust ducting, these items have a relatively low weight.

The gas-turbine engine, owing to its complete combustion process, does not produce toxic carbon monoxide as a combustion by-product as does the gasoline engine and, to a much lesser extent, the diesel engine. This is a desirable characteristic when the engine is installed in a vehicle where the operating personnel are restricted to a confined space. The gas turbine emits a a clean exhaust not visible to the naked eye, whereas the compression-ignition engine emits a smoke signature when the engine is accelerated.

Since the lubricating oil does not come in contact with the combustion products, the gas-turbine engine does not dilute, contaminate, or burn an appreciable amount of oil.

The gas-turbine engine has excellent low-temperature starting characteristics and requires a minimum of cold-starting aids. The turbine engine can start almost equally as well at -60°F as at +80°F within a 30-second period. The engine is able to operate at peak load almost immediately without the customary warm-up period. Although the engine itself will start in very low ambients without preheat aid, a problem still exists in obtaining sufficient battery voltage from cold batteries to rotate the turbine up to minimum self-sustaining speed. Once a turbine-powered vehicle is started, a large volume of high-temperature air is available for use as a prestart warm-up aid. Also, fuel that is not preheated must have a pour point below the existing ambient temperature.

The gas-turbine engine has the inherent ability to successfully burn all fuels within the military range specified as "multifuel." Fuel logistics are greatly simplified since it is not necessary to supply fuel with any specific octane or cetane number, but merely a hydrocarbon fuel whose characteristics satisfy, nonselectively, the characteristics of any of the fuels within the multifuel range.

Although the gas-turbine engine offers many apparent advantages as a vehicular power source, it does present certain disadvantages such as:

(a) The requirement for a large volume of air
(b) The requirement for a complex electrical system
(c) Poor overall fuel economy
(d) The generation of a high-frequency high-energy sound
(e) Heavy starting systems
(f) Component complexity
(g) Unproved reliability and life span
(h) High production cost.

The gas-turbine requires a large volume of air for combustion, approximately 3 to 4 times the quantity required by a compression-ignition engine. Higher pressure ratios and operating temperatures can reduce the air requirement to about twice that of the compression-ignition engine.

The internal components of the gas-turbine engine are vulnerable to particle erosion caused by the high rotating speed of the components and the high air velocity through the turbine. Thus efficient air filtration must be provided to protect the engine from the extremely dusty air that tracked vehicles generate during off-the-road operation. A filter that retains the separated dirt particles within the filter element cannot be used in a vehicle that encounters extremely heavy dust conditions because it soon would become clogged and ineffective. A filter unit, or more correctly, a separator unit that mechanically removes the dust particles from the air by either centrifugal or inertial force and ejects them overboard must be used. The gas-turbine engine is very sensitive to any depression of the absolute pressure of the intake air. The required filter components introduce a pressure drop in the air system, causing a decrease in output power.

The gas-turbine engine requires a much more complex electrical system than a conventional diesel engine owing to the incorporation of additional and necessary operating safety controls.

The gas-turbine engine has poorer overall fuel economy than a compression-ignition engine. Although the fuel economy of some present-technology regenerative-cycle gas-turbine engines, at rated speed and load, nearly equals the fuel economy of the compression-ignition engine, the low-load and idle-speed fuel consumption of the gas turbine is greater. Because military vehicles operate at idling and low-load conditions for long periods of time, more fuel will be consumed if the vehicle has a gas-turbine engine than will be used should the vehicle have a comparable compression-ignition engine.

The high-frequency high-energy sound generated by the gas-turbine engine, when unattenuated, can cause temporary and sometimes permanent damage to the hearing and sense of equilibrium of operating personnel. However, this problem can be resolved with proper design of inlet and exhaust silencers. These devices, like the air-filtration unit, introduce some loss of engine power.

The starting system of the smaller gas-turbine engine is heavy. For example, the complete starting system, including batteries for a 300-hp unregenerated turbine weighing 220 lb weighs 170 lb. A small single-shaft gas turbine, a small pulse-jet turbine, or a small free-piston gasifier, all of which operate on the same fuel as the main engine, could be used as a starting system. It is estimated that these starting units would weigh from 45 to 85 lb if designed to start a 300- to 500-hp gas-turbine engine. Cartridge starters have the lightest weight but are expensive. A standard Air Force cartridge designed to start a 300-hp turbine engine costs $20.

Simplicity, lower maintenance, and higher reliability have generally been considered to be advantages of the gas-turbine engine on the basis that it has fewer moving parts. This statement is generally true. However, this judgment is based on experience in aircraft and stationary installations, where the turbine is operating at near optimum conditions, i.e., constant speed, uniform loading, and installation in a high-rpm machine.

In a tracked or wheeled surface vehicle, the advantage of the smaller number of moving parts in a regenerated gas-turbine engine is offset by the complexity of those components and the increased complexity of the total system installation. In a wheeled or tracked vehicle, loads and speeds are constantly changing. Shock loads from the driving wheels or tracks, along with acceleration and deceleration, and shifting and steering loads, all of which are transmitted back to the turbine, cause detrimental temperature instabilities within the gasifier and power-producing sections and result in high thermal stresses, fatigue, and burning away of the wheel blades.

The gas-turbine engine has yet to demonstrate total superiority for any acceptable length of time over a comparable modern reciprocating compression-ignition engine when installed in an automotive vehicle subjected to the rigors of military operation. For most past applications the operating life of the turbine engine was less than 100 hr, owing to mechanical failures. Consequently at this time there is no accurate basis for stating that the gas-turbine engine has the advantages of simplicity, lower maintenance requirements, and high reliability compared with a compression-ignition engine installed in a military surface vehicle.

One must bear in mind that past and present installations of the gas-turbine engine as a prime mover have involved experimental engines installed in experimental vehicles. At time of installation the common view has been held that if volume production were required, refinements and advancements could be introduced at the time of such production. Further, the consensus has been that the total system would then in all probability prove equal or superior to present power-plant installations in surface vehicles.

The gas-turbine engine is not adaptable to production by existing piston-engine machine tools. Large-scale production tooling, necessary to reduce costs to a competitive level, will not be purchased by private industry until the turbine engine becomes commercially feasible. However, to be realistic,

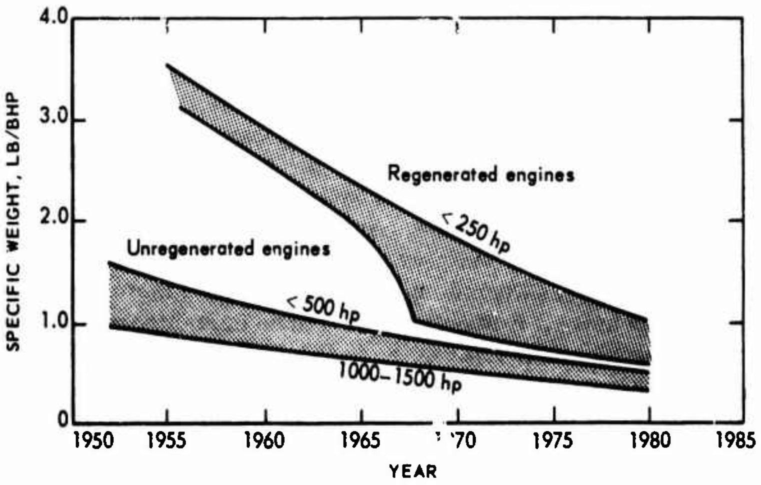


Fig. I-210—Trend Forecast of Specific Weight of Gas-Turbine Engines

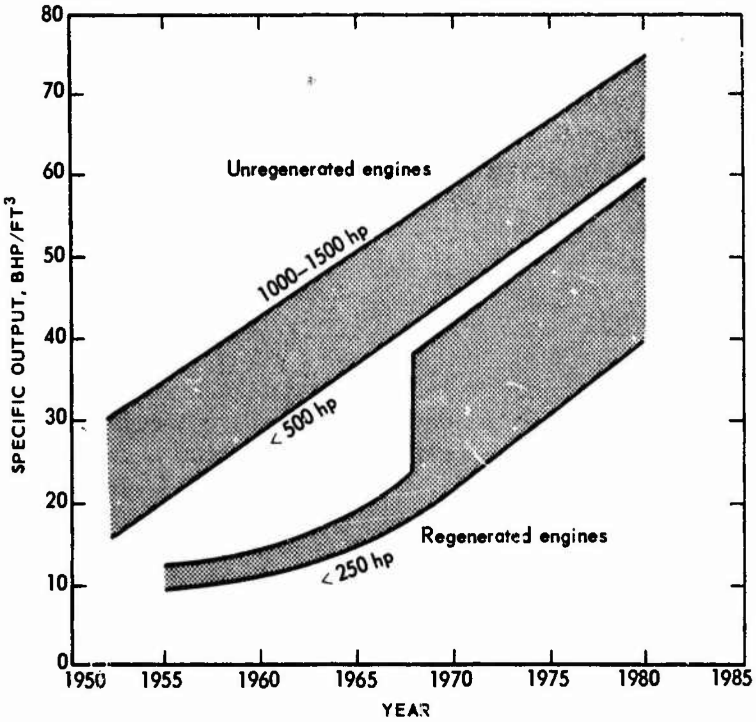


Fig. I-211—Trend Forecast of Specific Output of Gas-Turbine Engines

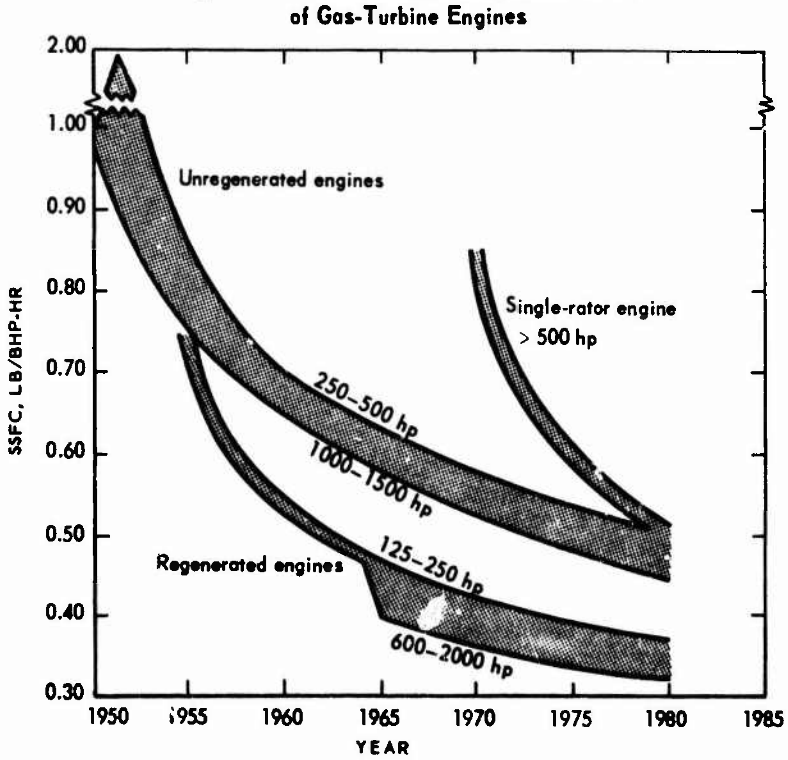


Fig. I-212—Trend Forecast of SSFC of Gas-Turbine Engines

it should be pointed out that the aircraft turbine engine was in the same predicament some years back. If the military had not funded production tooling, aircraft might not have the advantages of the aircraft gas turbine today.

The trend forecasts shown in Figs. I-210 to I-212 illustrate the estimated level of technological achievement anticipated for the gas-turbine engine through 1980.

CONCLUSIONS

At present no gas-turbine engines are applicable for use in tactical vehicles. Gas-turbine-engine technology is rapidly advancing, however, and indications are that the gas-turbine engine has the potential to improve the capabilities of certain tactical vehicles within the next 10 to 12 years. Vehicular application of the gas-turbine engine will first be found in commercial transcontinental trucks. Industry will pursue this development, since it is believed that there is a sufficient truck market to warrant producing these engines in production quantities. However, the power requirements for this market will not exceed 500 to 600 hp.

The operating environment for military tactical vehicles is much more severe than for commercial vehicles. Many of the advantages of a gas-turbine engine for commercial vehicles are lost when applied to tactical vehicles. However, some large tactical vehicles, such as the main battle tank, require very high power outputs from engines of minimum size and weight, and only the gas-turbine engine has the potential of meeting this requirement. Therefore, since industry has no incentive to develop gas-turbine engines in high power ranges, development of these engines must be sponsored by the Government. Government support of an R&D program could produce gas-turbine engines that would improve the physical and performance characteristics of some future tactical vehicles.

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Chapter 14

COMPOUND ENGINES

INTRODUCTION

The compound, or turbocompound, engine is essentially a 2- or 4-stroke-cycle reciprocating-piston engine incorporating a gas turbine driven by the engine's exhaust gases. The reciprocating engine and the gas turbine are arranged to provide power independently. The separate power outputs of each are transmitted through gearing to a common output shaft.

The compound-engine power concept differs from pure turbosupercharging where a turbine-compressor, driven by the engine exhaust gases, is used to boost charging pressure (manifold pressure charged to the engine cylinder above atmospheric pressure) in the engine. In the compound engine there are in essence two engines operating in series with the exhaust of the first engine boosting the horsepower ratio of the second engine. The power output then represents the combined individual outputs of each engine, with an increase in the output of the second engine (over the rated horsepower) resulting from the boosting effect of the exhaust gases from the first engine on the second engine. In the case of turbosupercharging, the power output represents only the boosted horsepower output of one engine.

TYPES OF COMPOUND ENGINES

Napier Nomad

The British Napier Nomad aircraft diesel engine, developed shortly after WWII, represented a significant advance in turbocompound engine development. This power plant was comprised of a 2-stroke-cycle loop-scavenged diesel engine, a gas turbine, and an axial-flow compressor, all coupled to form a complete energy-conversion device (see Fig. I-213).

The Napier Nomad power system provided extreme flexibility in the selection of operating conditions for best performance over a wide speed range. The engine was very light and compact for its power output and demonstrated a low SSFC of 0.36 lb/shp-hr at full power and a part-throttle SSFC of 0.325 lb/hp-hr. The performance characteristics of the Napier Nomad engine are presented in Fig. I-214.

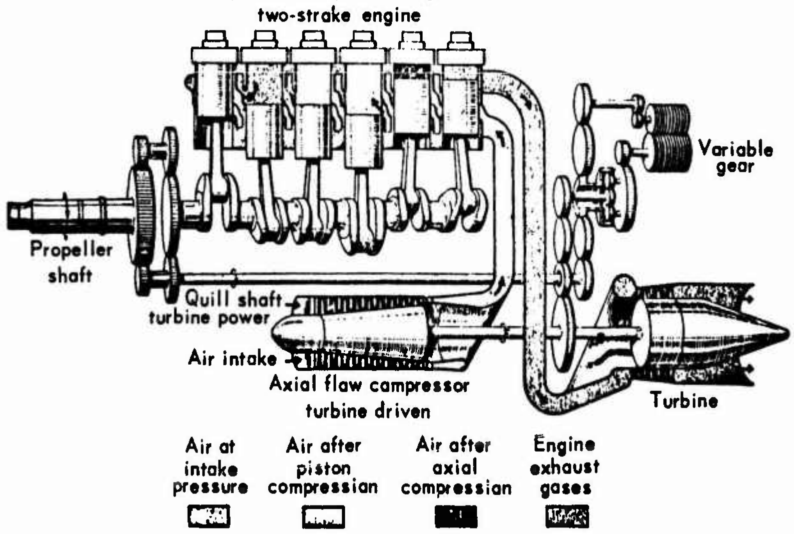


Fig. I-213—Schematic Arrangement, 3000-hp Napier Nomad Aircraft Turbocompound Diesel Engine

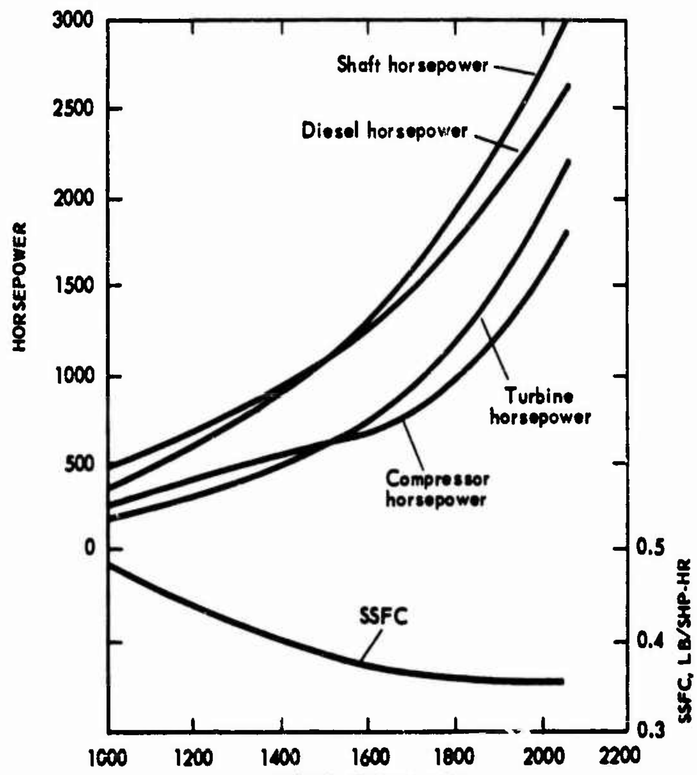


Fig. I-214—Performance Characteristics of Napier Nomad Aircraft Turbocompound Diesel Engine

ICAN atmosphere.

Curtiss-Wright and Allison Engines

More recent power plants that combine a piston engine with a gas turbine (or turbines) are the Curtiss-Wright Corporation's TC 18 and the Allison Division of GMC's V-1710-E27 (-127). The Curtiss-Wright TC 18, illustrated in Fig. I-215, is an improved version of the R-3350 2-row 18-cylinder 4-cycle air-cooled radial aircraft engine. The TC 18 incorporates three exhaust-driven gas turbines equally disposed around the engine. The power recovered from the engine exhaust gases by the turbines is fed back to the engine crankshaft by means of a gear train operating in conjunction with three fluid couplings. A 20 percent boost in engine power is achieved by this turbocompound system.

Allison's V-1710-E27 (-127), shown in Fig. I-216, is an improved version of the V-1710-G6, an Allison 12-cylinder 4-cycle liquid-cooled Vee-type aircraft engine. The V-1710-E27(-127) incorporates a single exhaust-driven gas turbine located at the rear of the engine. The power recovered from the engine exhaust gases is fed back to the engine crankshaft by means of a gear-reduction box. A 36 percent increase in power output and a 21 percent reduction in fuel consumption are achieved by this engine when operated at high altitude.

Project ORION

In 1950 the General Electric Company proposed a design concept to the military for an unusual compound engine and was subsequently awarded a contract for the development of a compound-cycle engine. Development of this engine was designated Project ORION. At the time this project was initiated it was anticipated that, if successful, the power plant would emerge as the ultimate engine for tracked-vehicle application, especially with respect to engine fuel economy and compactness.

The initial ORION project consisted of developing three similar small experimental engines. The compound-cycle engine of the ORION concept consisted of an air-cooled, reciprocating diesel-engine gas producer that drove a centrifugal compressor. The gases generated by the diesel gas producer were expanded in a blowdown puff-energy turbine and then fed through a constant-pressure turbine to produce shaft power. A schematic of this process is shown as Fig. I-217. Figure I-218 illustrates a cutaway view of an early experimental engine.

The final efforts of Project ORION were concentrated on developing a large tank engine called Rigel. The Rigel engine consisted of a supercharged regenerative air-cooled 2-stroke-cycle 6-cylinder opposed-piston diesel engine that drove two centrifugal compressors. One compressor supplied air to the combustion process, and the second compressor supplied air to cool the cylinders. The hot gases from the diesel combustion and the cylinder-cooling air were directed to the power turbine. A schematic diagram of this engine is shown as Fig. I-219, and a cutaway view of the experimental Rigel engine is shown as Fig. I-220.

At the outset it had been hoped that Project ORION would result in the development of a multifuel power source having a specific output of less than 2 lb/hp and greater than 10 hp/ft^3 and an SSFC of from 0.34 to 0.36 lb/shp-hr. However, during tests the SSFC of the 600-hp engine varied from 0.57 lb/shp-hr at full speed of 2350 rpm to 0.95 lb/shp-hr at a speed of 1500 rpm. At that

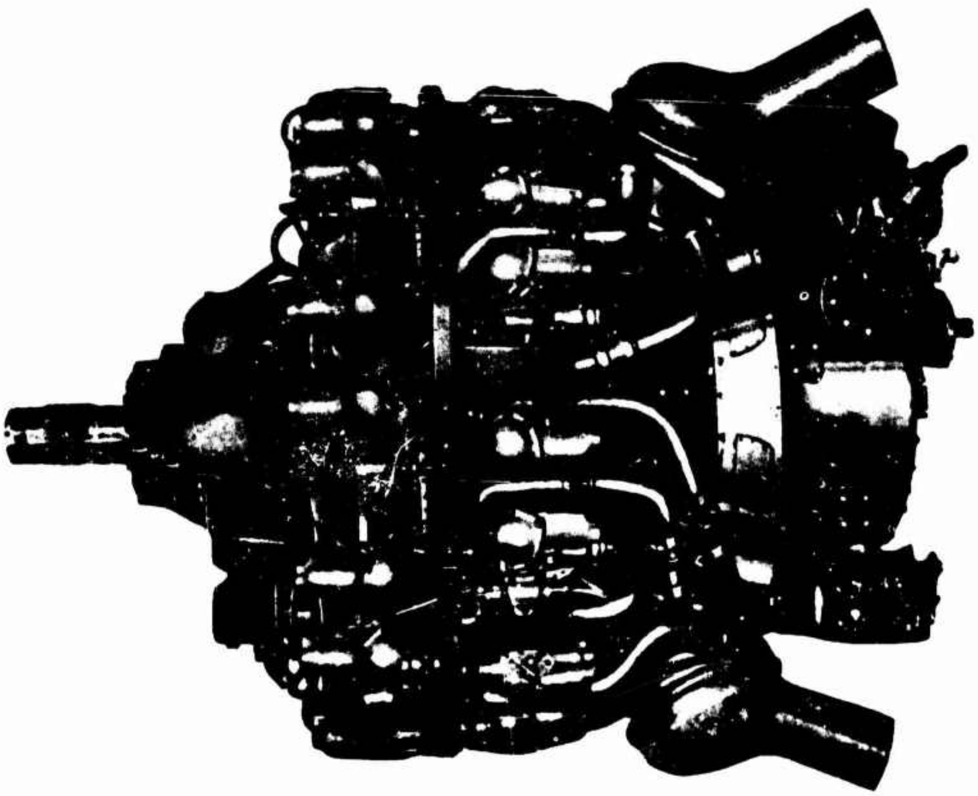


Fig. I-215—Curtiss-Wright TC 18 3400-hp Turbocompound Aircraft Engine

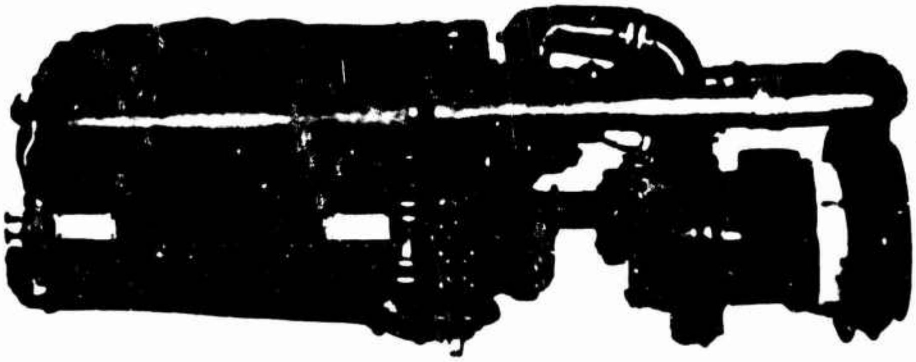


Fig. I-216—Allison Division V-1710-E27 (-127) Turbocompound Aircraft Engine

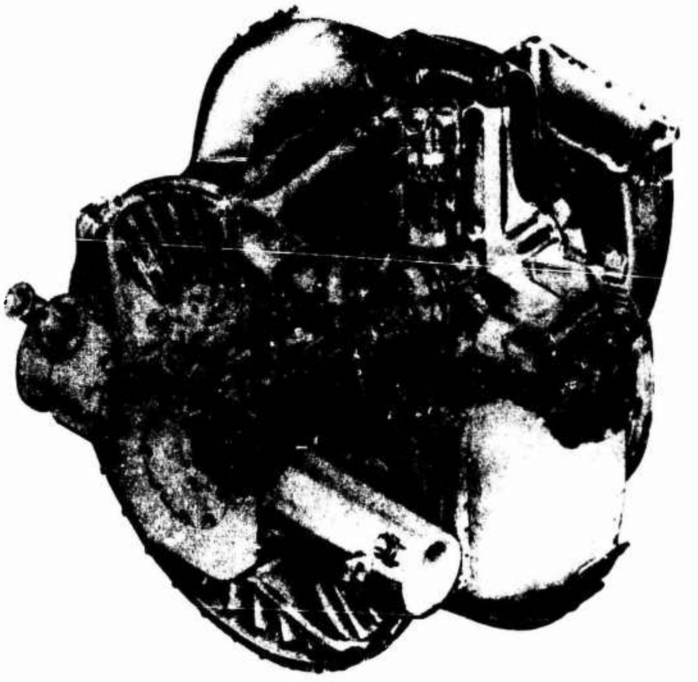


Fig. I-217—Schematic Diagram of Project ORION Experimental-Cycle Power Plant

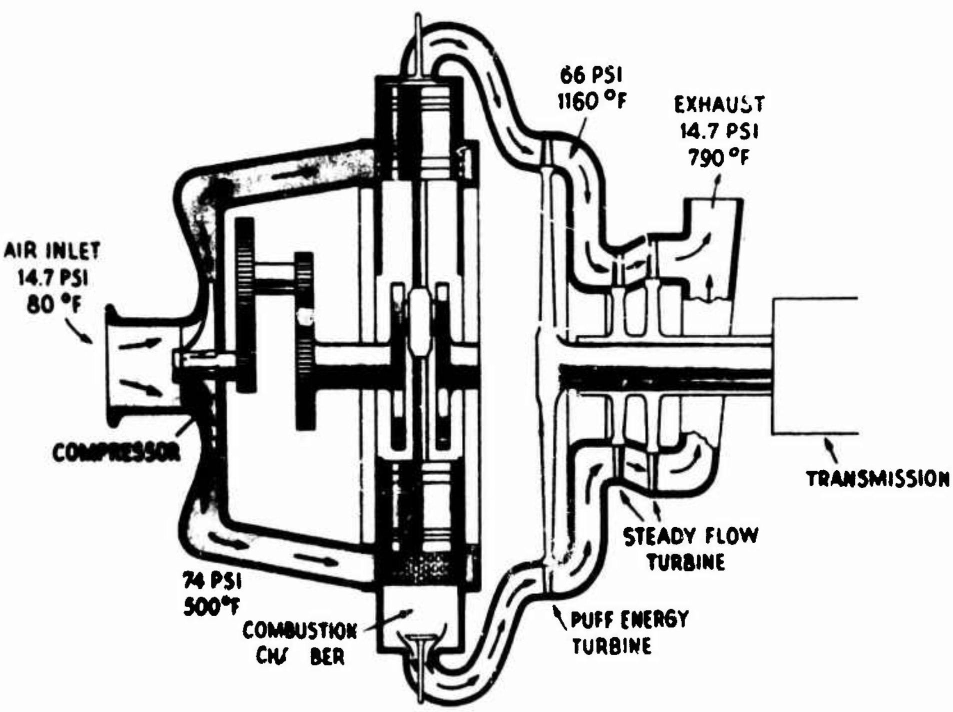


Fig. I-218—Cutaway View of Rigel Experimental Engine

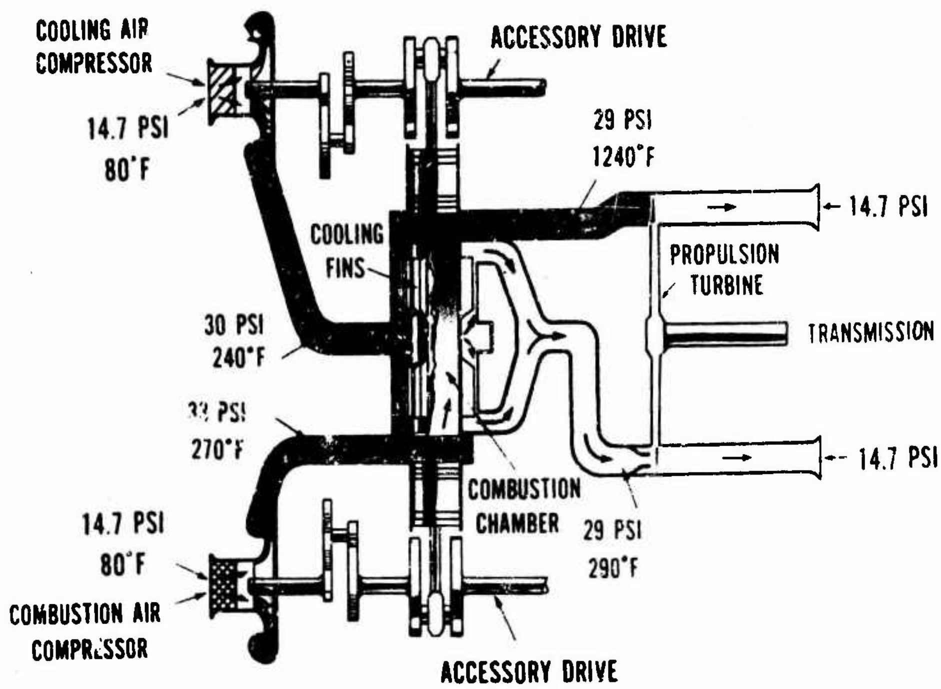


Fig. I-219—Schematic Diagram of Project ORION 600-hp Rigel Engine

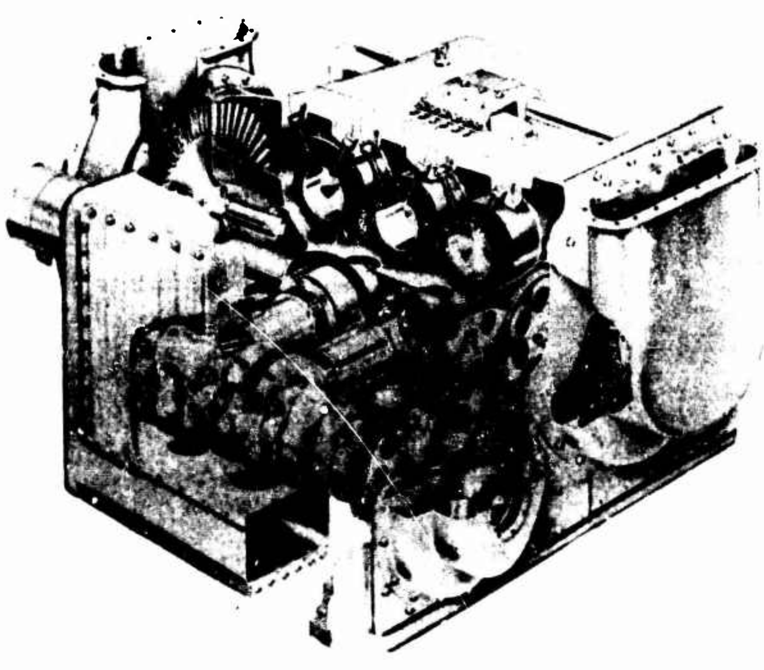


Fig. I-220—Cutaway View of Project ORION 600-hp Rigel Engine time it became apparent that a major redesign would be necessary to achieve specified goals. Even then there was serious doubt that the Rigel engine could be developed to meet the goals in order to compete with other power sources, i.e., the air-cooled diesel engine and the gas-turbine engine.

It is estimated that the Rigel compound engine, rated at 860 hp for continuous operation and 1400 hp for emergency, would have an SSFC of 0.419 lb/hp-hr and 0.446 lb/hp-hr respectively. A turbocharged compression-ignition engine of the same displacement as the gas-producing engine would develop 928 hp with an SSFC of 0.389 lb/hp-hr. Project ORION was terminated in the latter part of 1955.

Other Concepts

A recent Society of Automotive Engineers paper[1] discusses a unique combined-cycle engine arrangement. A schematic diagram of this engine is shown as Fig. I-221. The power source proposed uses a conventional 4-cycle compression-ignition engine with a "free" power turbine and a bypass line from

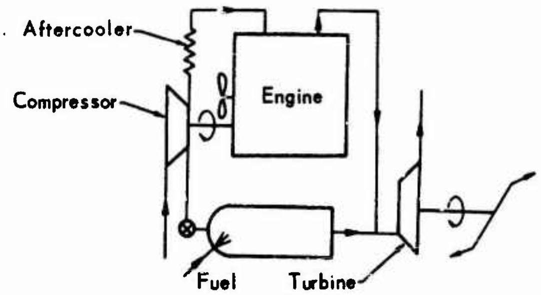


Fig. I-221—Compound-Engine Turbine as a Power Generator with Engine Bypass

the compressor to a burner and thence to the gas turbine. The configuration permits equality between the reciprocating engine and the compressor and turbine, and the required balance between these components is maintained. A small fraction of the air passes through the engine and absorbs heat in the process. The balance of the air required for the turbine is directed to the combustion chamber, where additional fuel can be injected as required. These gases then flow to the turbine and mix with the exhaust gases from the reciprocating engine.

Figure I-222 is a schematic diagram illustrating a simple compound engine in which the power from the turbine is fed back to the crankshaft. Figure I-223 is a schematic diagram illustrating a simple compound engine which utilizes the gas produced in the generator to power the free turbine. The Project ORION engines operated on this principle.

Variable-geometry nozzles for the turbine of the compound engine that would improve engine acceleration have been evaluated. A torque almost four times that of conventional engines is obtained. Figure I-224 compares the relative torque curves of a conventional engine and a compound engine, each having

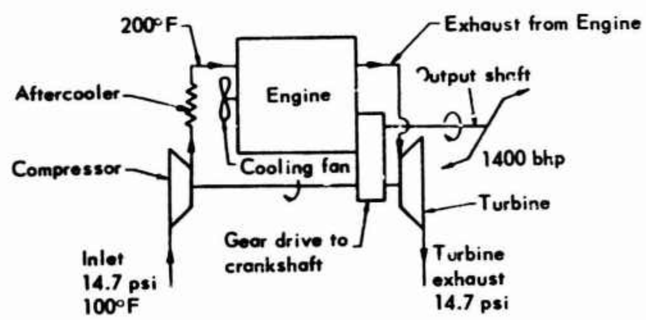


Fig. I-222—Simple Compound Engine with Turbine-Power Feedback to Crankshaft

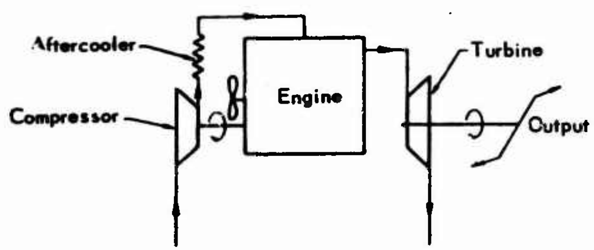


Fig. I-223—Simple Compound Engine with Engine as Gas Generator Supplying Turbine Output

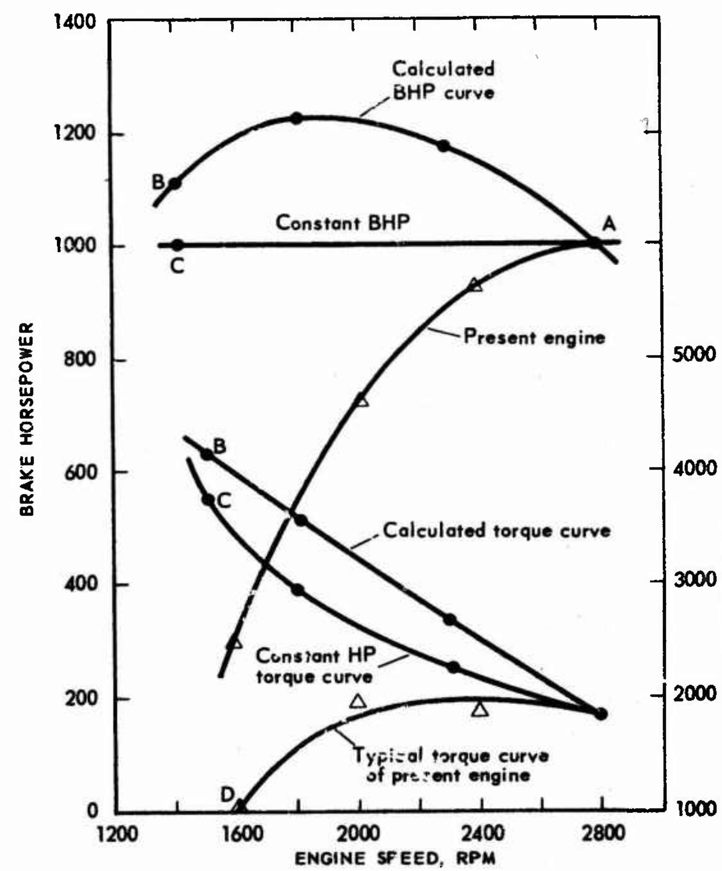


Fig. I-224—Relative Torque Curves of Conventional and Compound Engines with Variable-Geometry Turbine Nozzles a variable-geometry-nozzle turbine. The SAE paper[1] points out that a turbine having a variable-geometry nozzle is ". . .thought to be a requirement in any case for any major advancements in turbocharging. . .[or turbocompounding capabilities], . . .and is even required for the simple compound engine at low speeds."

Piston-Turbine Compound Engine

The most recent attempt to develop a compound engine has been made by the Southwest Research Institute. The engine, designated a piston-turbine compound engine (PTC), is shown schematically in Fig. 1-225. The PTC engine is a uniflow 2-stroke-cycle opposed-piston piston-supercharged turbocompound diesel engine with VCR. It obtains high efficiency by operating at a high temperature and pressure.

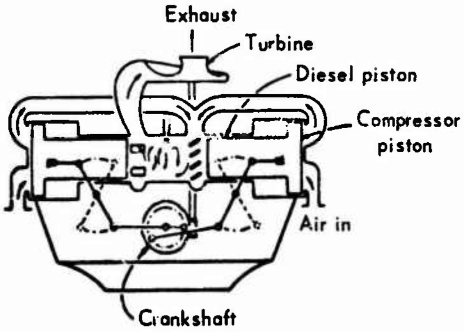


Fig. I-225—Schematic Diagram of PTC Engine

In 1961 the Southwest Research Institute initiated a study to investigate the PTC engine. The analysis indicated that the engine had great potential for military applications. The study concluded that such an engine would be light and compact, with low fuel consumption and a true multifuel capability.[2]

ATAC was interested in the PTC engine concept since it had the potential to provide tactical-vehicle engines with excellent physical and performance characteristics. After a favorable evaluation of the design analysis, the military supported a computer study to optimize the basic PTC engine design and operating cycle. At time of writing this phase is being completed.

A cutaway view of the basic PTC engine design concept is shown in Fig. I-226. A large reciprocating compressor piston attached directly to, or made an integral part of, the power piston accomplishes scavenging and supercharging. The reciprocating motion of the diesel power pistons is converted into circular motion by rocker arms. One end of each rocker arm is attached to, and oscillates with, a piston. The opposite end of the arm is attached, through a connecting rod, to the crankshaft that holds a central position in the engine.

The rocker-arm fulcrum is eccentrically mounted on a fulcrum shaft, enabling the compression ratio of the diesel engine, or the pressure ratio of the compressor, to be varied by controlled rotation of the fulcrum shaft. The mechanism

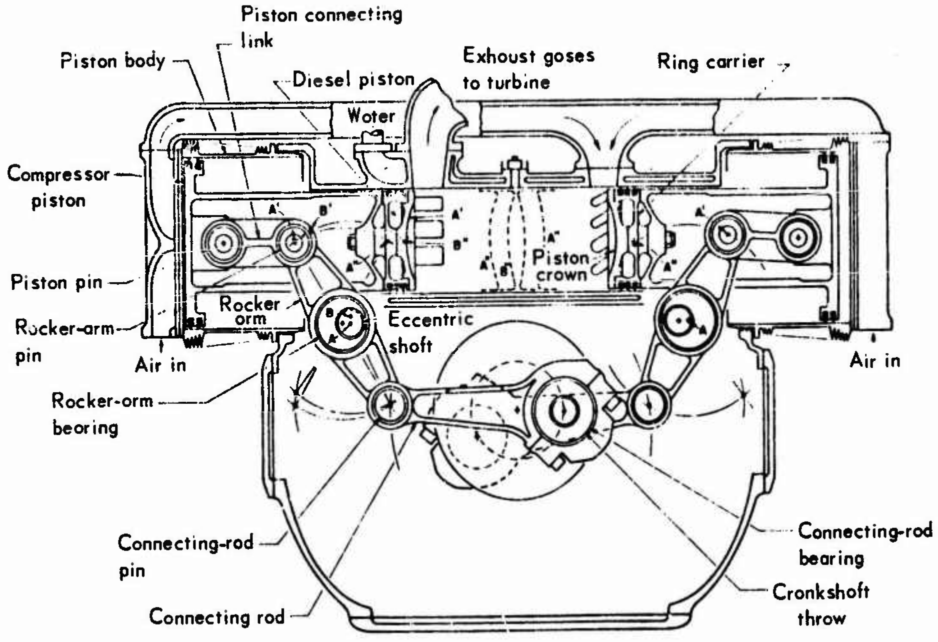


Fig. I-226—Cross Section of PTC Engine

by which VCR is achieved is shown in Fig. I-227. The ability to vary the compression ratio of the diesel and compressor cylinders is attained by mounting the rocker-arm bearings eccentrically on shafts that protrude through the side of the engine block. Levers are mounted to the shafts through which the eccentric shaft can be rotated while the engine is in operation. The levers can be actuated during the engine-start cycle (by a remote start-mode switch) and scheduled automatically as a function of engine speed during the load-running cycle, thereby maintaining a constant compression pressure as engine speed varies. This can be accomplished by means of a pneumatic, hydraulic, or electrical device, pressure-controlled from a source in the engine. When the eccentric shaft is rotated, the center of the rocker-arm bearing changes position, changing the inner and outer dead points of the piston stroke. The linkage mechanism is designed so that the total stroke length remains essentially unchanged. The piston inner and outer dead points move an equal distance in the same direction for a given rotation of the eccentric shaft.

Similar VCR devices have been demonstrated with success. A series of pressure-actuated reed valves on the compressor and diesel inlet ports is utilized.

In PTC engine operation, the compressor pistons discharge compressed air into the combustion chamber of the diesel power cylinder through a series of ports at one end of the cylinder. Combustion gases are exhausted through a similar series of ports at the opposite end of the cylinder. A radial-inflow turbine is connected to the exhaust ports by ducting to utilize exhaust-gas blowdown energy. The turbine is located in close proximity to the power cylinders. The power generated by the gas turbine is transmitted to the engine crankshaft by means of gearing. An overrunning clutch is positioned between the gear and crankshaft to avoid feedback from the diesel to the turbine during acceleration.

Many conventional diesel engines are referred to as having a "multifuel" capability if another heavy fuel, in addition to No. 2 diesel fuel, can be used for successful engine operation. However, a power source having true multifuel capability should burn a wide range of fuels, i.e., gasoline, diesel, CITE,

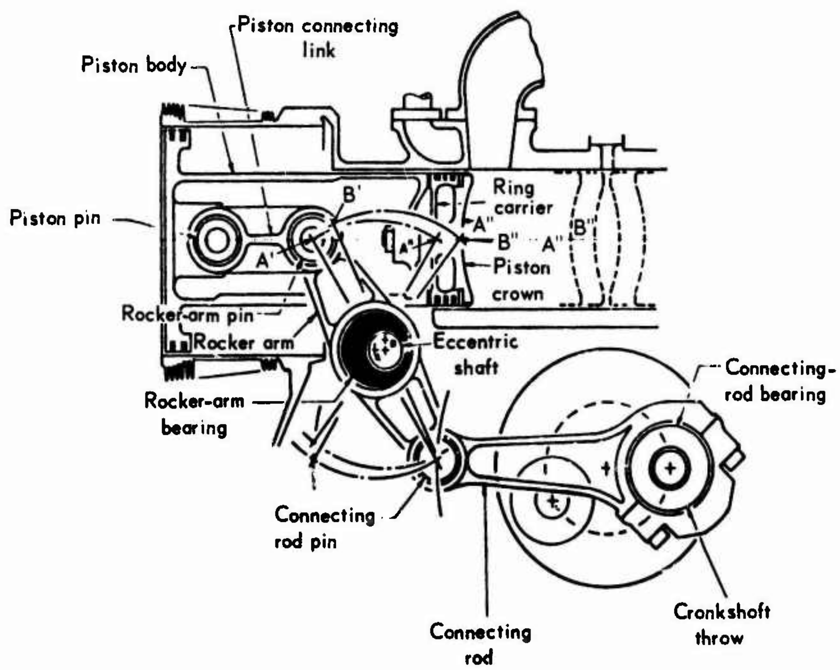
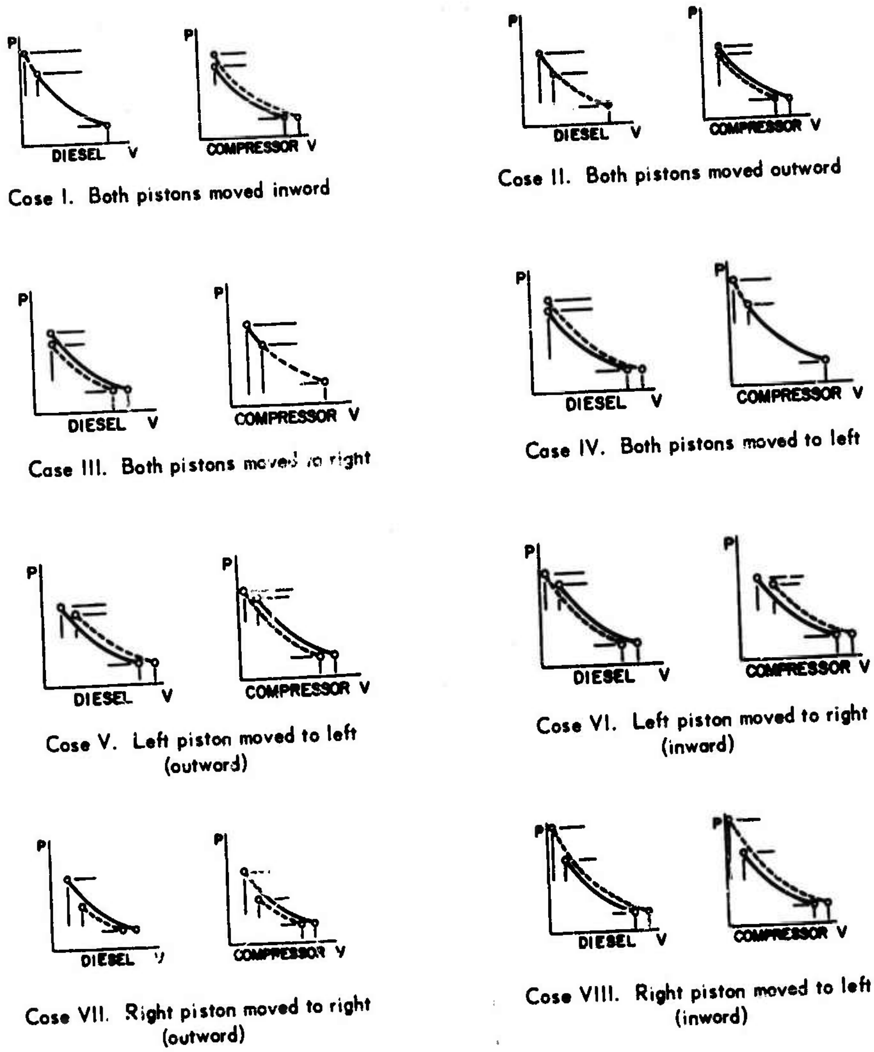


Fig. I-227—Variable-Compression-Ratio Mechanism, PTC Engine

JP-4, etc., under a wide range of climatic conditions. To achieve this capability, the engine should have a variable compression ratio to maintain a reasonable pressure level and long life.[3]

A power source with a VCR offers versatility for noncritical fuel requirements, cycle-pressure control, and short-duration high-power bursts. Many diesel engines that are reported to have a multifuel capability operate at extremely high ratios and pressures.

High compression ratios are required only to start and operate the PTC engine at low speeds. The pressure-volume diagrams of Fig. I-228 illustrate the effects that can be obtained by the VCR system of the PTC engine.



Case I. Both pistons moved inward

Case II. Both pistons moved outward

Case III. Both pistons moved to right

Case IV. Both pistons moved to left

Case V. Left piston moved to left (outward)

Case VI. Left piston moved to right (inward)

Case VII. Right piston moved to right (outward)

Case VIII. Right piston moved to left (inward)

Fig. I-228—Pressure-Volume Diagrams of PTC Engine: Effect of VCR

The greatest changes in compression ratio of the diesel and compression cylinders are controlled by movement of the right piston only (as in directional change), as shown in cases VII and VIII of the pressure-volume diagrams. This result indicates that the timing of the opening and closing of the intake ports is of considerable importance. It is therefore desirable to vary the compression ratio of only the right-hand piston assembly and leave the left-hand piston assembly mounted in conventional fashion without a control device. The PTC engine has a wide-range multifuel capability in that it is possible to raise the compression pressure sufficiently to start and operate this engine on gasoline and to lower the compression pressure while the engine is burning diesel fuel. The proper control of pressure gives added life to the engine.

DISCUSSION

Compound engines are overly complex, and therefore no serious effort is being made in this country to develop compound engines, with the exception of the PTC-engine-development program now in progress at the Southwest Research Institute.

The PTC engine is in the design-concept stage, and hardware has not been produced to date (1966). However, the PTC concept appears to be feasible and practical, with the desired characteristics for a tactical vehicle.

The PTC engine has inherent flexibility in design configuration. Although a minimum of two cylinders must be used for proper matching of the turbine to assure a continuous gas flow to the turbine, any number of cylinder banks, such as two, three, four, or six, may be stacked to form a complete unit. This feature enables the PTC engine to have good family capabilities. Possible cylinder arrangements of the PTC engine are illustrated in Fig. I-229. Stacking more than six cylinders is impractical owing to the need of a long crankshaft.

Figure I-230 illustrates the general arrangement of a proposed 3-cylinder engine, 4-in. diesel bore size, with an estimated continuous rating of 500 hp.

Table I-22 illustrates the family capabilities of the PTC engine for small (3-in.) and large (4-in.) bore sizes. The table lists both the continuous and intermittent power ratings. For continuous power output the diesel air-fuel ratio is 28 to 1. This can be lowered to 21 to 1 for intermittent duty, which increases the power output by approximately 50 percent. Figure I-231 illustrates the wide range of power obtainable from the family of engines with two bore sizes.

PTC engine cooling is accomplished both by a liquid-cooling and an air-cooling system. The diesel section is liquid-cooled. The compressor section and manifold section are air-cooled by a fan integral to the engine. The use of a radiator is required to reject heat in the liquid system. It appears that cooling the diesel section by air will be difficult, although not impossible.

For the most part the PTC engine can be manufactured using present automotive-engine tooling since the power-producing components are similar to the reciprocating components of a conventional automotive engine. The turbine portion can also be manufactured without difficulty. The turbine portion of the engine operates at a maximum inlet temperature of 1385°F. This temperature is from 300 to 400° cooler than the temperature of a conventional gas-turbine engine. Consequently the turbine can be manufactured from readily available low-cost alloys.

Figure I-232 illustrates the estimated fuel-power performance characteristics of a 4-in.-bore 2-cylinder PTC engine. The normal continuous rating of this engine is 330 hp. Additional PTC engine specifications are shown in Table I-23. Figure I-233 shows the estimated maximum- and reduced-power

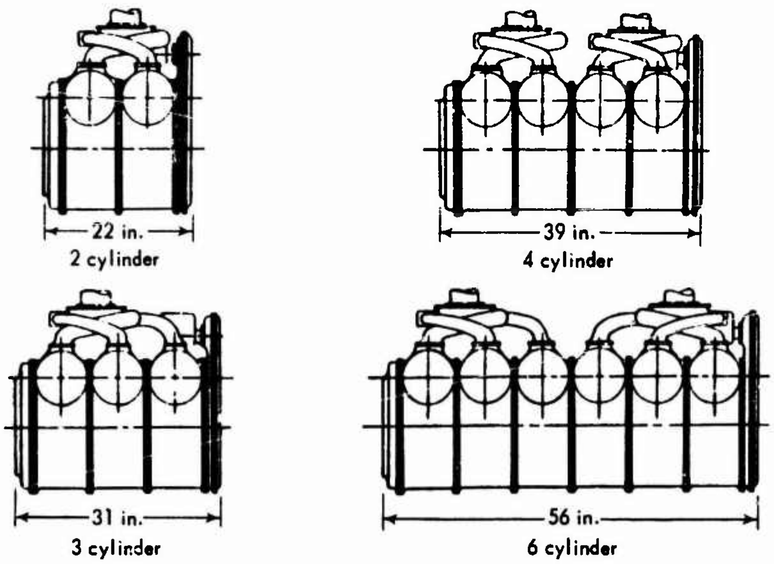


Fig. I-229—Possible Cylinder Arrangements and Family Capabilities of PTC Engine

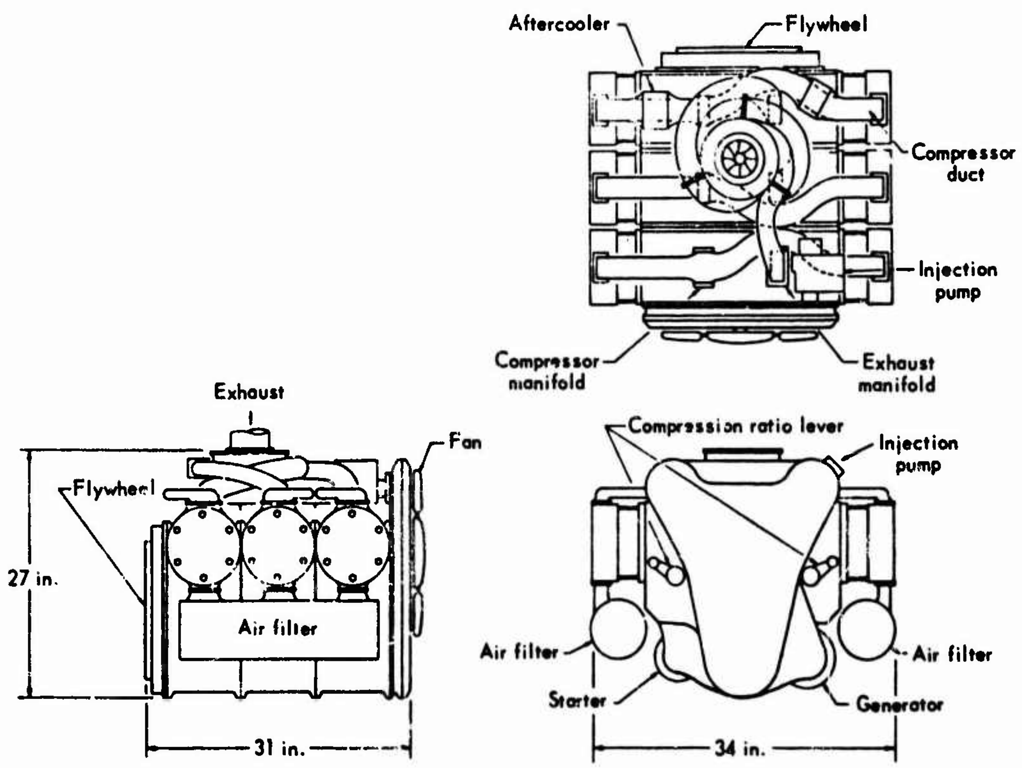


Fig. I-230—PTC Engine Configuration, 3-Cylinder Version

TABLE I-22

Family Capabilities of Piston-Turbine Compound Engine

No. of cylinders	Power rating, hp	
	Small-bore diesel (3-in. bore)	Large-bore diesel (4-in. bore)
Two		
Continuous	150	330
Intermittent	220	500
Three		
Continuous	225	500
Intermittent	340	750
Four		
Continuous	300	660
Intermittent	450	900
Six		
Continuous	450	1000
Intermittent	680	1500

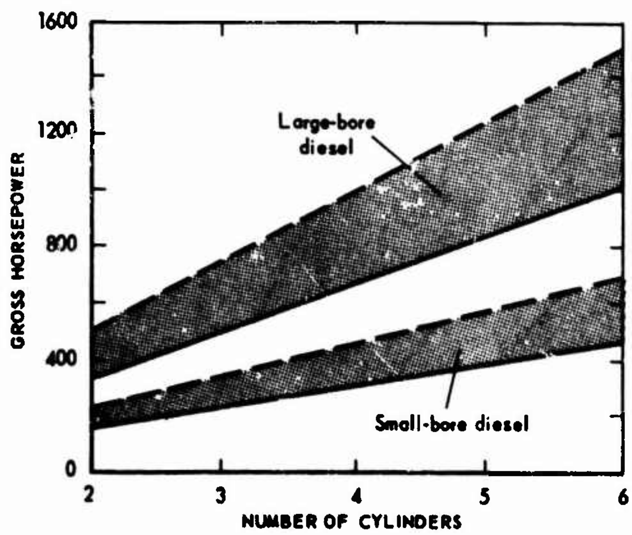


Fig. I-231—Family Capabilities of PTC Engine

Intermittent rating — Continuous rating

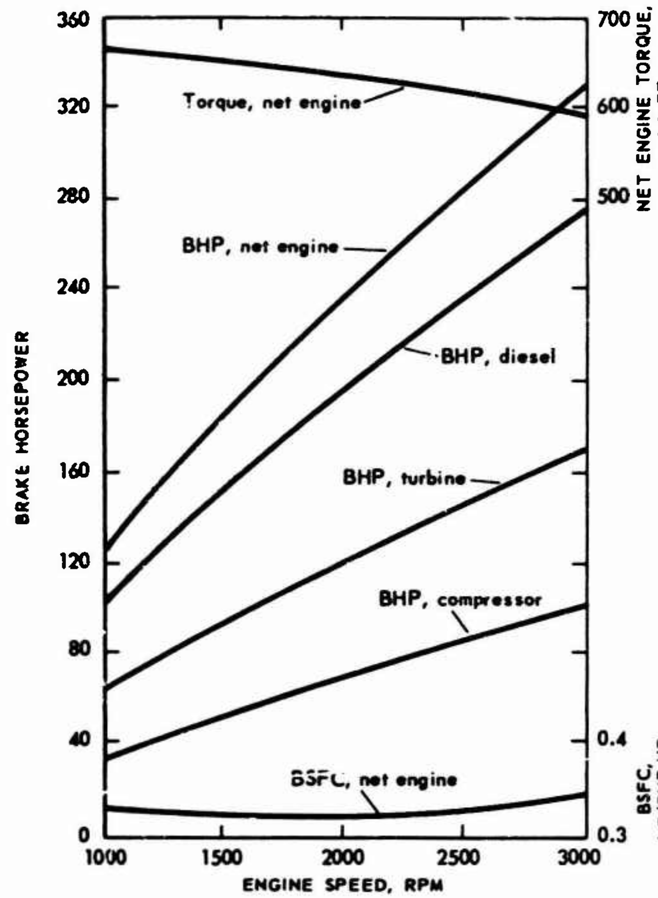


Fig. I-232—Estimated Full-Power Performance of PTC Engine
4-in.-bore diesel, two cylinders.

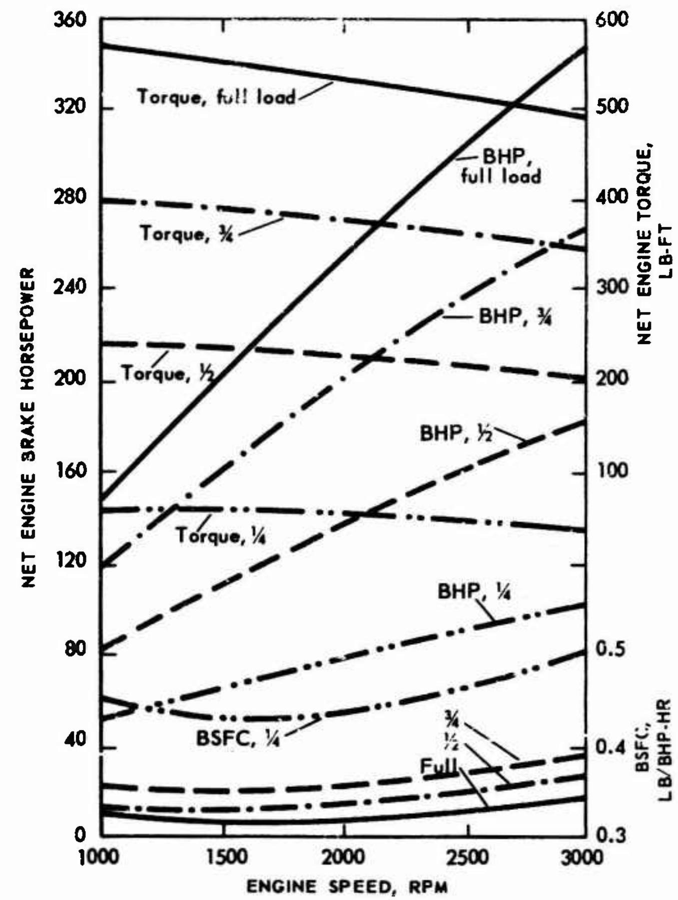


Fig. I-233—Estimated Maximum- and Reduced-Power Performance of PTC Engine
4-in.-bore diesel, two cylinders.

performance characteristics of this unit. The fuel consumption curves show a minimum SSFC of 0.32 lb/hp-hr at 1500 rpm and 0.35 lb/hp-hr at full load. In this horsepower range, conventional compression-ignition engines have a minimum SSFC of 0.38 to 0.40 lb/hp-hr.

TABLE 1-23

Specifications of PTC Engine

(4-in.-bore diesel)

Item	Value or description
Diesel	
Bore, in.	4.0
Stroke, 2 × 4, in.	8.0
Displacement, one cylinder, $in.^3$	105
Maximum engine speed, rpm	3000
Piston speed, fpm	2100
Compression ratio	Variable
Combustion chamber	Direct
Cooling, diesel cylinder	Water
Compressor	
Bore, in.	7.0
Stroke, 2 × 4, in.	8.4
Displacement, two pistons, $in.^3$	323
Intake valve	Reed-type
Exhaust valve	Reed-type
Airflow (at 83% vol. eff.), lb/min	33.5
Pressure ratio	3.75 : 1
Cooling	Air
Turbine	
Type	Radial-inflow
Maximum turbine-inlet temperature, °F	1385
Maximum turbine-inlet pressure, psia	55
Maximum speed, rpm	55,000

The horsepower and torque characteristics of the PTC engine are comparable to those of a conventional diesel engine having similar power and speed characteristics. The predicted performance of the PTC engine is based on the assumption that:

(a) The unit will have a compressor volumetric efficiency of 83 percent.

(b) The "short-circuit" air during scavenging will be 23 percent of the compressor-delivered air.

(c) The diesel cycle is the limiting pressure cycle with injection, designed to hold a 500-psi maximum pressure rise during combustion.

(d) Maximum power in the diesel section is restricted to a fuel-air ratio of 60 percent of stoichiometric to control smoking in the manifold and turbine.

(e) The turbine efficiency varies from 80 to 70 percent, depending on speed and load. Actual recovery of energy in exhaust gas at the exhaust-port opening is approximately 43 percent.

The Southwest Research Institute, which is developing a 4-in.-bore 6-cylinder 1000-hp PTC engine, estimates that the engine will weigh approximately 3000 lb in ferrous construction and occupy 30 ft^3. These characteristics correspond to a specific power output of 3.0 lb/hp and 33 hp/ft^3. However, if the engine were constructed of nonferrous material, such as aluminum or magnesium, the weight could be reduced to approximately 2400 lb, or 2.4 lb/hp. The weight and volume characteristics of the PTC family of engines, based on present technology, are listed in Table I-24.

TABLE I-24

Weight and Volume Characteristics of PTC Engine[a]

Item	Engine horsepower			
	330	500	660	1000
	Value			
No. of cylinders	2	3	4	6
Weight, lb (est.)	880[b]	1300[b]	1650[b]	2400[b]
Dimensions, L × W × H, in. (approx.)	22 × 34 × 27	31 × 34 × 37	39 × 34 × 37	56 × 34 × 37
Volume, ft^3	12	17	21	30
Specific weight, lb/hp	2.7	2.6	2.5	2.4
Specific output, hp/ft^3	27	29	31	33

[a]Data based on present technology.
[b]Nonferrous construction.

The PTC engine has not yet been built and would require approximately 3½ to 4½ years of development before units could be available for tactical vehicles.

CONCLUSIONS

Compound engines appear to be feasible and practical and able to operate at low fuel consumption. This characteristic makes these engines applicable for tactical vehicles where low fuel consumption is of greater importance than engine size and weight.

The inherent characteristics of the compound engine make it more suited for military than commercial vehicle application. Because of this, commercial engine manufacturers are not interested in developing compound engines with their own funds. The military would therefore have to sponsor the development of the compound engine.

The future potential of the compound engine, including the PTC engine, could result in an engine with (a) reduced fuel consumption, (b) multifuel capability, and (c) increased power output without an increase in weight or size.

Figures I-234 and I-235 forecast the level of technological achievement for the compound engine through 19

It is concluded that an R&D program sponsored by the Government would produce a compound engine that could improve the physical and performance characteristics of many future tactical vehicles. However, the development of compound engines sponsored by the Government for use as gas generators to supply gas to a free-turbine power output is not warranted at this time.

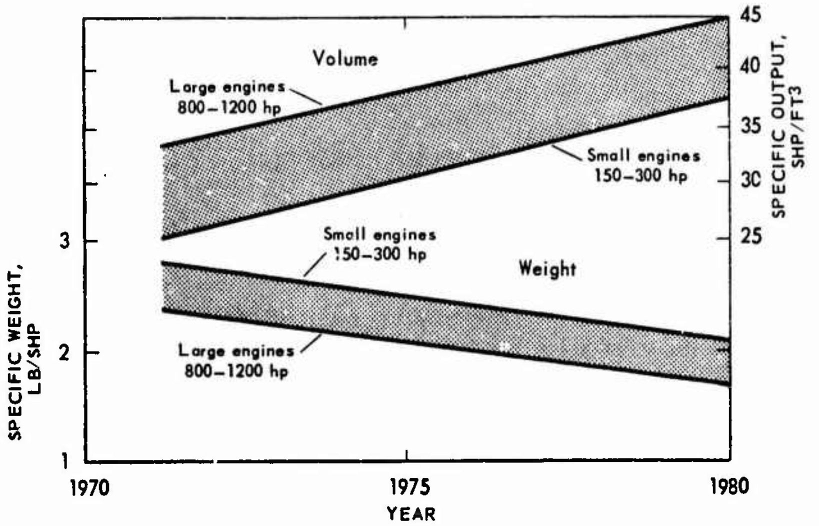


Fig. I-234—Forecast of Specific Weight and Specific Output of PTC Engine

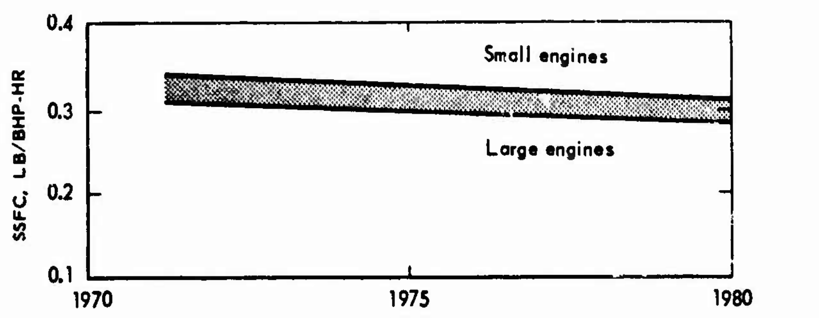


Fig. I-235—Forecast of SSFC of PTC Engine

REFERENCES

1. E. T. Vincent, "The Modern Compound Engine," SAE Paper 660131, Jan 66.
2. Southwest Research Institute, "The Piston-Turbine-Compound Engine," PTC Report, 14 Jun 66.
3. Julius E. Witzky, Ross F. Meriwether, and Floyd B. Lux, "Piston-Turbine-Compound Engine—A Design and Performance Analysis," SAE Paper 650632, Aug 65.

Chapter 15

FREE-PISTON ENGINES

INTRODUCTION

The basic concept of the free-piston engine dates back to the early 1920's when Pescara first proposed this unique power source. The basic objectives of this proposal were to develop a low-cost, lightweight, and compact power source with high efficiency that could operate on a wide range of fuel, independent of octane or cetane ratings, and that possessed torque characteristics superior to those of conventional reciprocating engines. However, many problems were encountered during the early phases of development, and consequently very little further effort was made to develop this engine concept until recent years.

In the early 1950's the French company SIGMA pursued development of the original Pescara design and, together with its licensees, has produced and sold approximately 600 of these gasifier (gas-generator) units throughout the world. Large industrial free-piston gasifiers have found application in electrical generating plants, compressor plants, pumping stations, dredges, ships, and locomotives. The SIGMA GS-34 gasifier, shown in Fig. I-236, is an inward-compressing machine capable of delivering 1230 gas hp.

The French company S. N. Marep has recently introduced the Model EPL-H40 gasifier, shown in Fig. I-237, which is an outward-compressing gasifier capable of delivering 2300 gas hp or 2000 turbine shaft hp.

Both of these French free-piston-turbine machines operate on heavy Bunker-C oil and have an SSFC of approximately 0.40 lb/shp-hr and an overall efficiency of approximately 36 percent. The units are heavily constructed, durable industrial machines not applicable to automotive vehicles.

The Baldwin-Lima-Hamilton Company developed an 800-shp free-piston engine for the US Navy Bureau of Ships that was intended as a power source for small-ship propulsion. The development program had reached the prototype hardware stage when terminated. Although some design deficiencies were experienced, the unit demonstrated satisfactory overall performance.

Many other large companies throughout the world have investigated and conducted R&D programs for this unique power source, but no engines have succeeded in being accepted for commercial use. Interest in this engine concept was displayed in the mid-1950's by several automotive companies, who developed and built limited numbers of these units and installed them in test

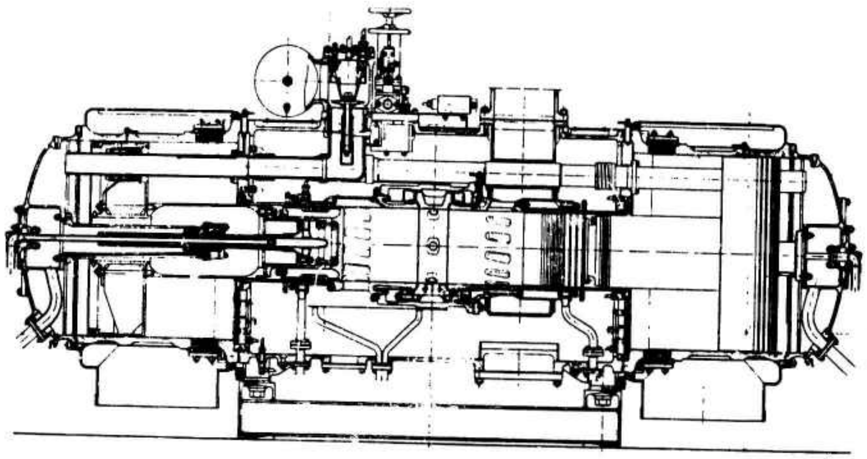


Fig. I-236—Cross Section of SIGMA GS-34 Gasifier Unit

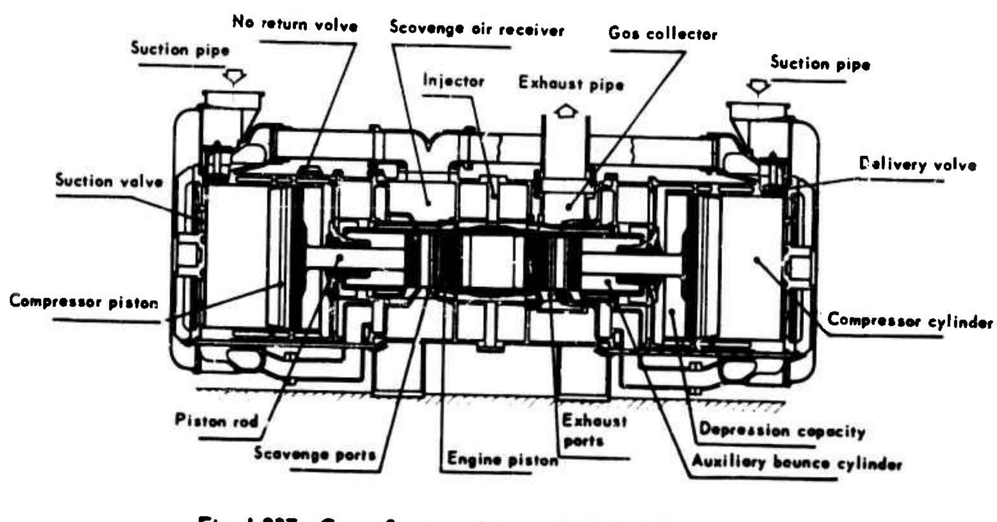


Fig. I-237—Cross Section of Marep EPL-H40 Gasifier Unit automobiles and farm tractors. However, they too discontinued their development work. A small recently established Canadian research organization is perhaps the most active in the development of the free-piston engine today.

PRINCIPLES OF OPERATION

The free-piston engine is basically a free-piston gasifier or gas generator that supplies hot gases to a turbine wheel. Figure I-238 is a schematic diagram of a basic gasifier turbine engine. The gasifier can be considered as an opposed-piston uniflow 2-stroke-cycle compression-ignition engine. The power pistons are rigidly connected to large compressor pistons. One side of the compressor-piston chamber serves as the gas generator and the other side serves as a bounce chamber to return the power piston through the compression stroke. The compression ratio is automatically varied as the power-load

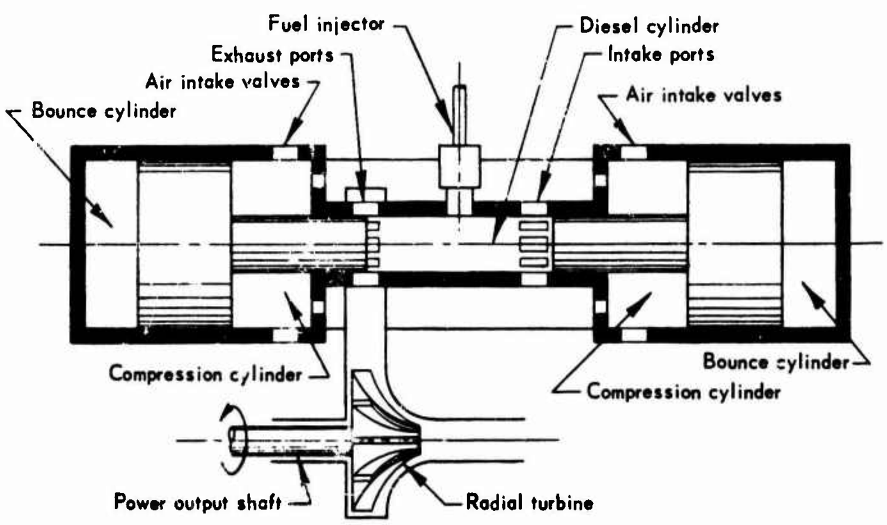
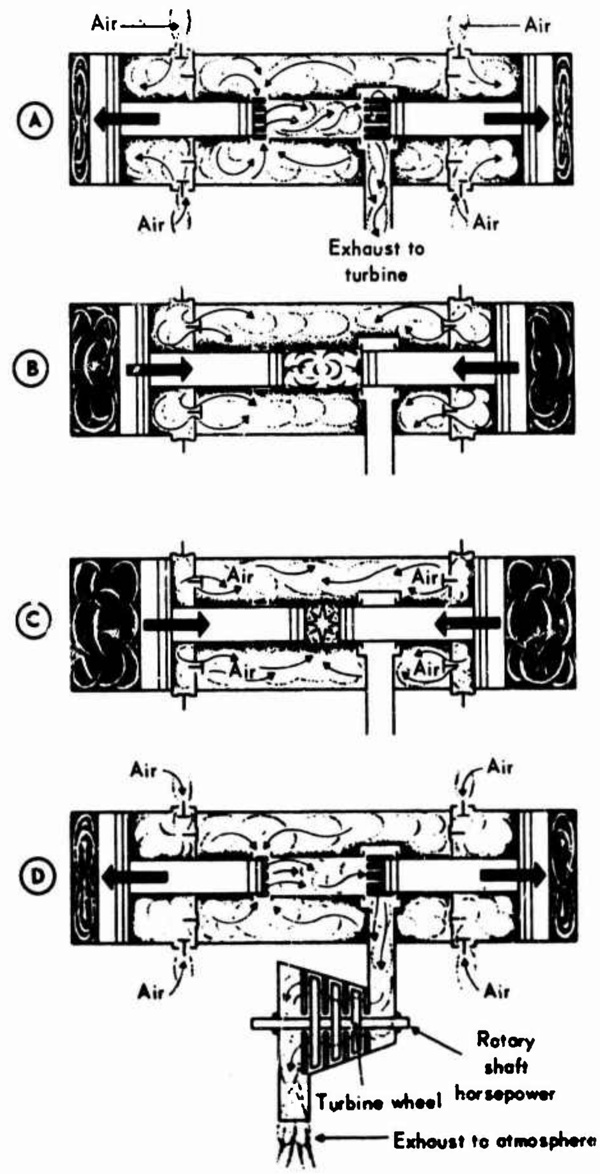


Fig. I-238—Schematic Diagram of Free-Piston-Gasifier Turbine Engine

demand is varied, thereby enabling these units to operate at optimum compression and combustion pressures. Figure I-239 illustrates the principles of operation of the free-piston-gasifier turbine engine.

There are two basic types of free-piston gasifiers: inward-compressing and outward-compressing. Figure I-240 schematically illustrates both types.

The outward-compressing gasifier has a compressor piston mounted coaxially to the power piston. Both units are in work position at the opposite end of the stroke. The pistons are driven by air compressed in the bounce cylinder during the previous stroke to the inner dead point of their respective chambers (equivalent to top-dead-center), where the injected fuel is ignited by compression. The pistons are forced outward by the combustion reaction on the diesel cylinder, which compresses the air within the compressor chamber. The compressed air is utilized to scavenge the diesel chamber and to drive the turbine wheel after passing through the combustion cycle. A change from part to full load is accomplished by throttling.



In Figure A, the pistons are completing their travel outward and are about to bounce back. Intake and exhaust ports are open and intake air is filling the compressor cylinder.

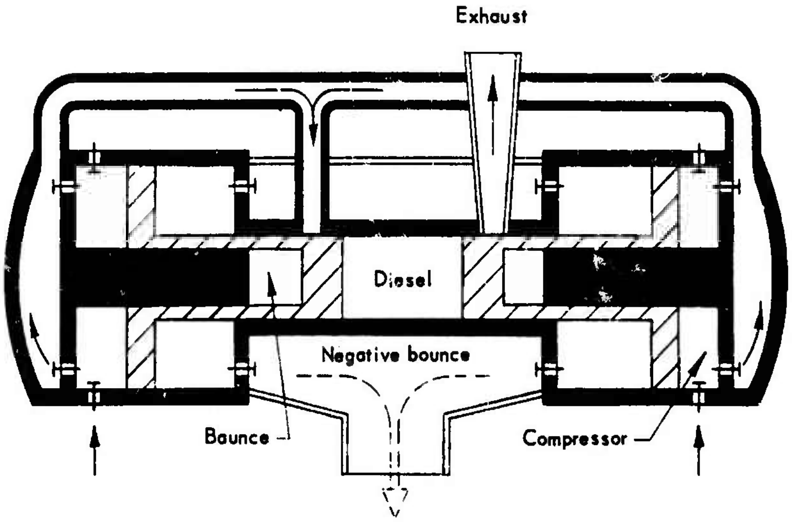
In Figure B, the pistons travel inward pumping air from the compressor cylinder into the air box, trapping air in the Diesel combustion space. Intake and exhaust ports are closed—air delivery valves are open.

In Figure C, pistons are completing inward travel. Fuel is injected into cylinder. This is combustion or the beginning of the power stroke. Intake and exhaust ports are still closed—air delivery valves are open.

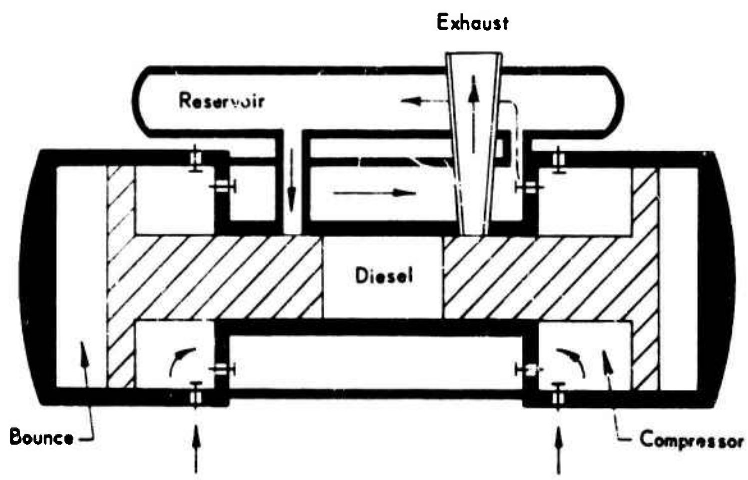
Figure D, shows the end of power stroke, compressing air in bounce space to return pistons for next cycle. Exhaust and intake ports are just opening to scavenge Diesel cylinder. Exhaust gases escape to turbine, spinning turbine wheels for usable power. Air is being drawn into the compression chamber.

Fig. I-239—Principles of Operation of Free-Piston-Gasifier Turbine Engine

The inward-compressing gasifier has a compressor piston coaxially mounted to the power piston. Both units are in work position at the same end of the stroke, i.e., the inner dead point of their respective chambers. The



a. Outward-Compressing Gasifier



b. Inward-Compressing Gasifier

Fig. I-240—Schematic Comparison of Inward- and Outward-Compressing Gasifiers bounce chamber on the opposite side of the compression chamber returns the piston through the combustion and compressor strokes. The inward-compressing gasifier requires the use of an air reservoir since air is compressed at a time in the cycle when exhaust scavenging is not required. A change from part to full load requires the use of a bypass or blowoff valve at the low-power-load end to maintain an ideal compression ratio for efficient operation.

Table I-25 compares the advantages and disadvantages of the inward-compressing and outward-compressing gasifiers.

TABLE I-25

Comparison of Advantages and Disadvantages of Inward-Compressing and Outward-Compressing Gasifiers

Inward-compressing gasifier	Outward-compressing gasifier
Better full-load fuel economy, but poorer part-load fuel economy	Better part-load fuel economy, but poorer full-load fuel economy
Operates at higher part-load (idle) frequency (has less speed shift from full load to minimum idle speed)	Operates at lower part-load (idle) frequency, which results in less frictional wear
Requires complicated full-load range controls (blowoff)	Requires simple controls to cover the entire load range
Requires compressor reservoir	Requires separate bounce cylinder, which results in greater friction loss
Simpler and more rigid construction possible	Better accessibility of central section
Less friction loss since separate bounce cylinder is not required	Good thermal insulation between cold intake and hot exhaust
Shorter gasifier unit	Longer gasifier owing to arrangement scheme
Engine cannot "carry over" should a misfire occur	Engine can "carry over" should a misfire occur
Lower efficiency due to greater pumping loss	Efficiency 2 to 4% greater owing to lower pumping loss
	Requires external air-transfer ducts

Both the inward- and outward-compressing gasifiers require some type of synchronous restraint of the piston assemblies within a common cylinder or in multicylinder units to maintain their phase relation. The mechanism to accomplish this can be either a toggle linkage or a rack and pinion.

The unique feature of the free-piston engine is that the compression ratio is automatically varied as the power and load demand is varied. This enables the gasifier to operate at optimum compression and combustion pressure cycles for efficient burning of almost any type of fuel at a given power level.

Free-piston gasifiers operate at very high compression ratios. Small automotive-type free-piston gasifiers operate at compression ratios from 40:1 to 50:1. Consequently very high pressures are generated during the compression and combustion cycles. VCR offers versatility inasmuch as it enables the engine to burn a wide range of fuels such as gasoline, diesel, CITE, JP-4, etc.

Starting of free-piston gasifiers is accomplished by compressed air forced into the bounce chambers. The compressed air can be supplied by air bottles or by a small engine-driven compressor supplying an air tank.

Free-piston engines are relatively free from vibration during operation owing to symmetrical piston motion. The gasifiers are very quick to respond to acceleration owing to the low inertia of the reciprocating components. The driving of accessories by free-piston turbine engines is difficult owing to the absence of a rotating drive on the gasifier and the difficulty of driving off the power turbine, which may be at stall or near-stall speed during certain periods of vehicle operation. An acceptable method of providing a drive for accessories would be to incorporate a separate low-pressure low-speed exhaust turbine located downstream of the main power turbine. A second method would be to drive the accessories by an electric motor powered by a high-speed alternator driven by a high-speed turbine located upstream of the main turbine.

DISCUSSION

The free-piston engine is in the early R&D stage, and no automotive-type units have been produced for acceptable application.

The free-piston turbine engine has inherent design flexibility for vehicle configuration and installation. Most recent gasifiers (with the exception of the GMC GMR-4-4 Hyprex) incorporate a single-cylinder configuration. Figure I-241 illustrates a cross-sectional view of the Ford 519 free-piston turbine engine, which is of the inward-compressing type.[1] This unit is a single-cylinder 3.75-in.-bore diesel that has developed approximately 160 gas hp and 120 shp. The external configuration of the Ford 519 gasifier is shown in Fig. I-242. Figure I-243 illustrates the arrangement of its reed-type air-inlet valves. Figure I-244 is a cross section of the Free Piston Development Company, Ltd., Series 5000 single-cylinder 3.5-in.-bore diesel gasifier of the outward-compressing type that develops approximately 100 gas hp. The series 5000 gasifier is presently under development by the Free Piston Development Co., Ltd., which has built and sold a small number of these units for experimental use. The external configuration of the series 5000 gasifier is shown in Fig. I-245. Figure I-246 illustrates a cross section of the GMC GMR-4-4 Hyprex free-piston gasifier, which consists of two cylinders "twinned" together in a common assembly. This unit is of the inward-compressing type with a diesel bore of 4 in. and develops approximately 360 gas hp. The twin-cylinder configuration resulted in a gasifier-unit length approximately 20 in. shorter, although somewhat wider, than if it were designed as a large-bore single-cylinder unit. The external configuration of the GMR-4-4 gasifier unit is shown in Fig. I-247. Free-piston gasifiers may be designed with either one, two, three, or four cylinders. The general arrangement of a proposed 4-cylinder outward-compressing free-piston turbine engine that develops approximately 600 hp is shown in Fig. I-248. In multicylinder arrangements the individual cylinders are phased to each other to provide a more continuous flow of gas; i.e., the cylinders of a 2-cylinder unit are phased 180 deg to each other, and in a 3-cylinder unit, they are phased 120 deg apart.

The free-piston turbine engine can be built as a unit with the gasifier and turbine forming a common assembly, or the turbine unit may be placed remotely from the gasifier unit with a gas duct connecting the two. This flexibility allows many possibilities for optimum vehicle installation. Figure I-249 illustrates

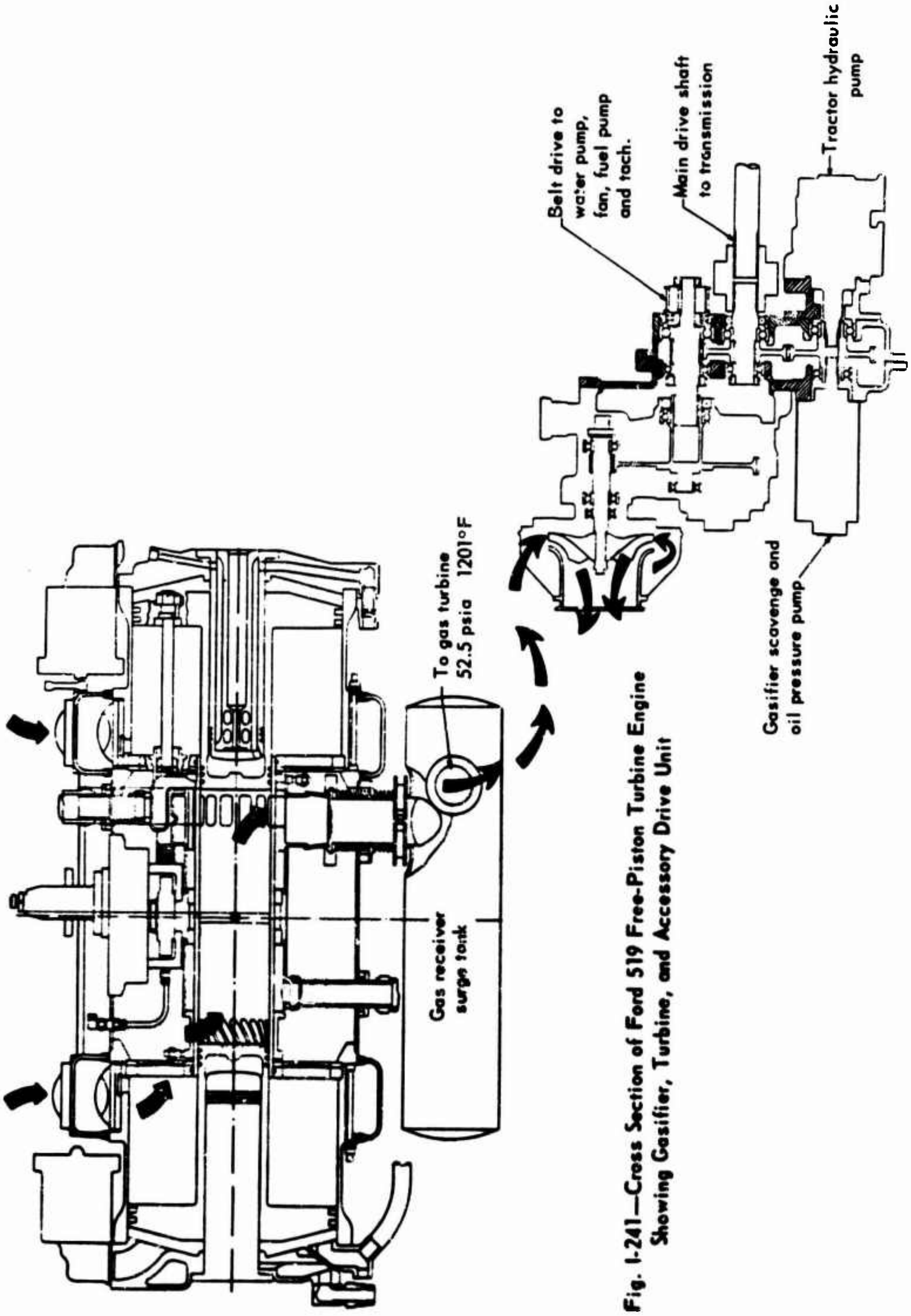


Fig. I-241—Cross Section of Ford 519 Free-Piston Turbine Engine Showing Gasifier, Turbine, and Accessory Drive Unit

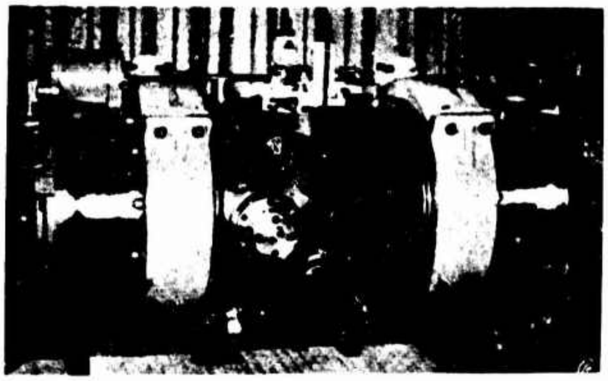


Fig. I-242—Ford 519 Free-Piston Gasifier

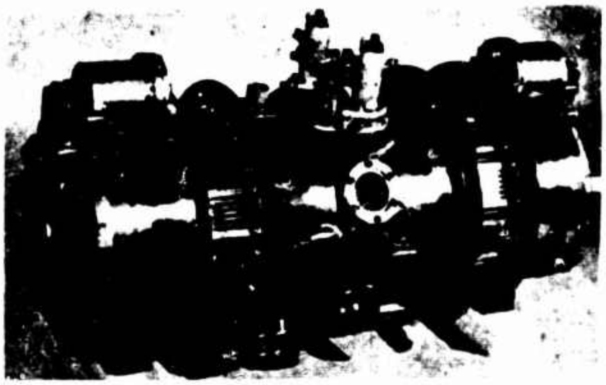


Fig. I-243—Ford 519 Gasifier with Intake Manifolds
Removed Showing Reed-Type Air-Inlet Valves

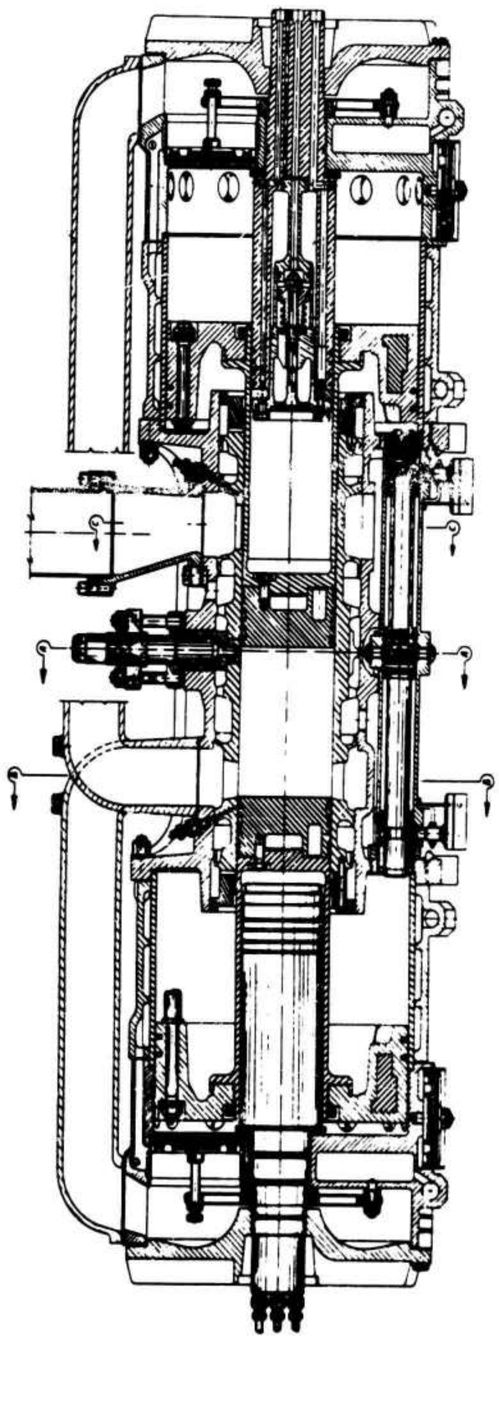


Fig. I-244—Cross Section of Free Piston Development Co., Ltd., Series 5000 Free-Piston Gasifier Unit

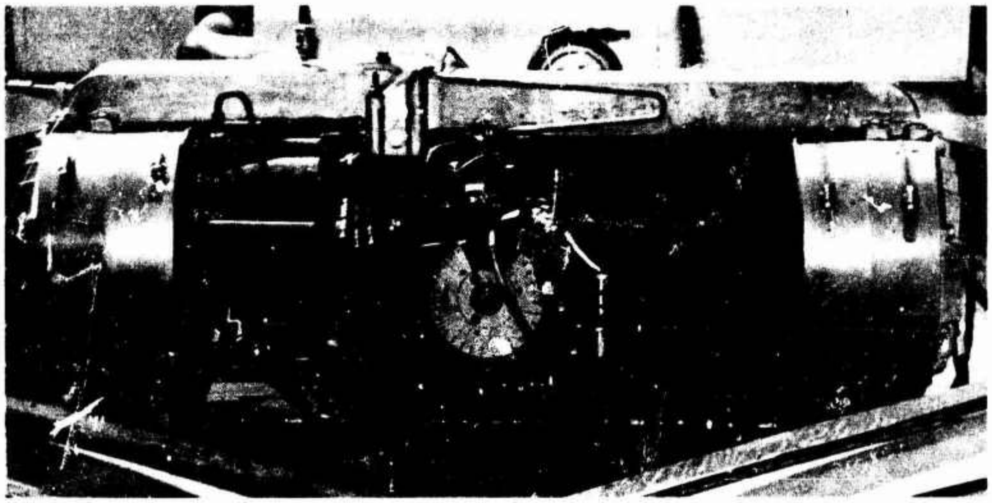


Fig. I-245—Series 5000 Gasifier Unit

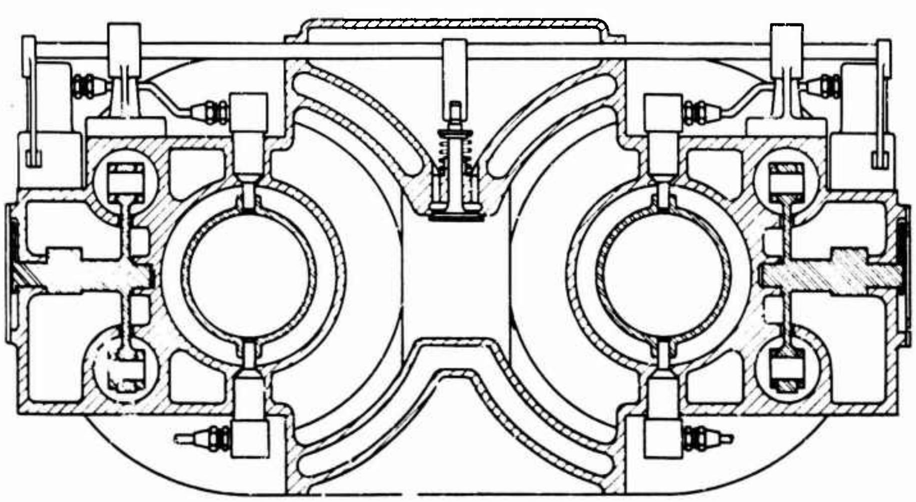


Fig. I-246—Cross Section of GMC GMR-4-4 Hyprex Siamesed
Twin-Cylinder Free-Piston Gasifier

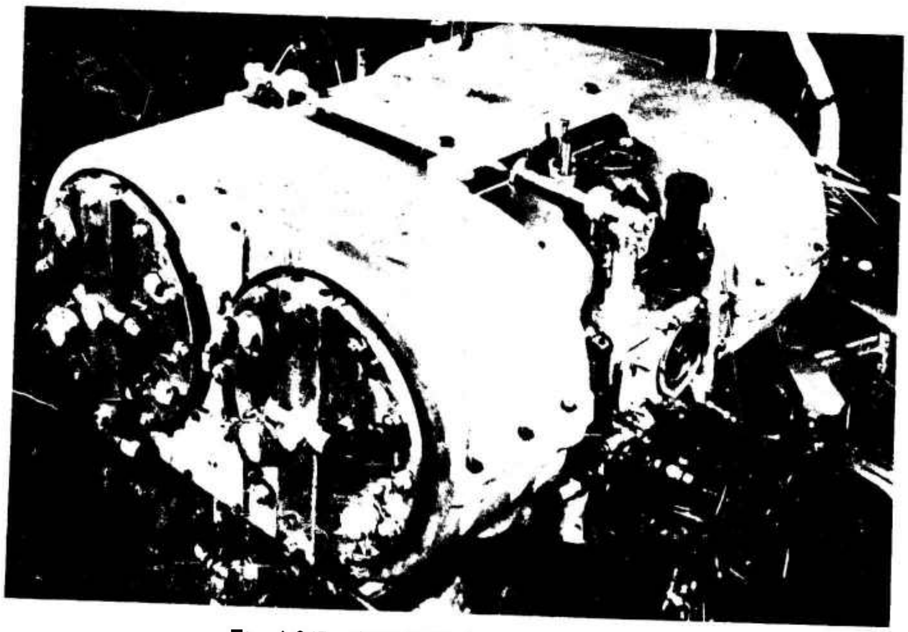


Fig. I-247—GMC GMR-4-4 Hyprex Gasifier

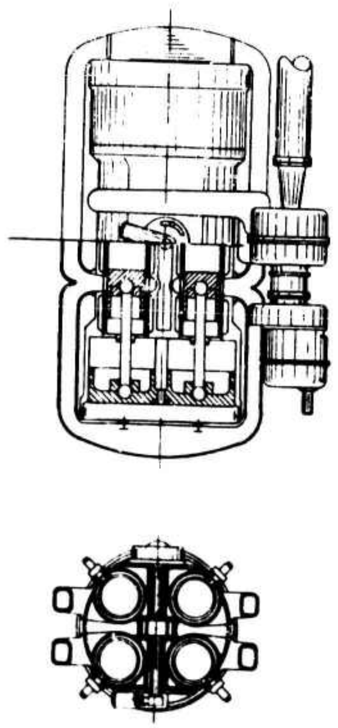


Fig. I-248—General Arrangement of Possible 600-hp
Multicylinder Free-Piston Turbine Engine
with Close-Coupled Gas Turbine

a free-piston turbine engine installed in an automotive vehicle with the gas-turbine transmission section mounted integrally to the engine. A propeller shaft connects the transmission output to the differential input, in the same

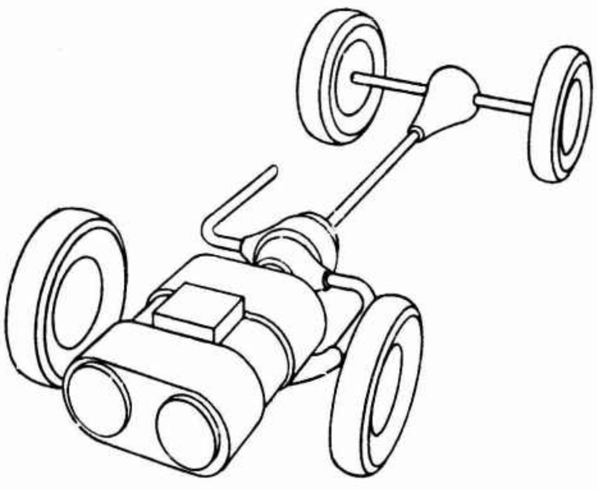


Fig. I-249—Schematic Diagram of Integral Gasifier-Turbine Unit in Automotive Vehicle

manner as with a conventional power source. Figure I-250 illustrates a gas-turbine transmission-differential that is located remotely from the gasifier at the driving axle of the vehicle. A duct connects the gas turbine to the gasifier unit. Figure I-251 illustrates a proposed 150-shp free-piston turbine engine with an integral transmission. The turbine, reduction gearing, and transmission are mounted axially to the cylinder center line. These units may also be mounted transversely to the cylinder center line. Figures I-252 and I-253 illustrate a free-piston engine installed in a farm tractor. It was reported[1] that the engine improved the performance of the tractor compared to that with a conventional engine and permitted the tractor to operate at increased loads without changing gear ratios. Figure I-254 illustrates the same gasifier unit, coupled with a gas turbine and gear box, installed transversely in a passenger vehicle.

Cooling of free-piston engines can be accomplished by utilizing a liquid or air cooling system. The diesel section of the gasifier is liquid-cooled, and the compressor section can be either liquid- or air-cooled. In vehicle applications in which the gasifier section is installed in a confined area where there is little natural air flow, a small cooling fan is utilized. In most applications, however, the fan used for cooling the liquid-system radiator may also be utilized to circulate air past the compressor section of the gasifier. Because of the high pressures and temperatures generated in the diesel section, it appears that air cooling of this section is difficult.

Free-piston turbine engines, for the most part, can be manufactured by using present automotive-engine tooling since the power-producing components are similar to the reciprocating components of a conventional automotive engine. Since the turbine operates at inlet temperatures 300 to 500°F lower than those of a conventional gas-turbine engine, the turbine wheel can be manufactured of readily available low-cost alloys.

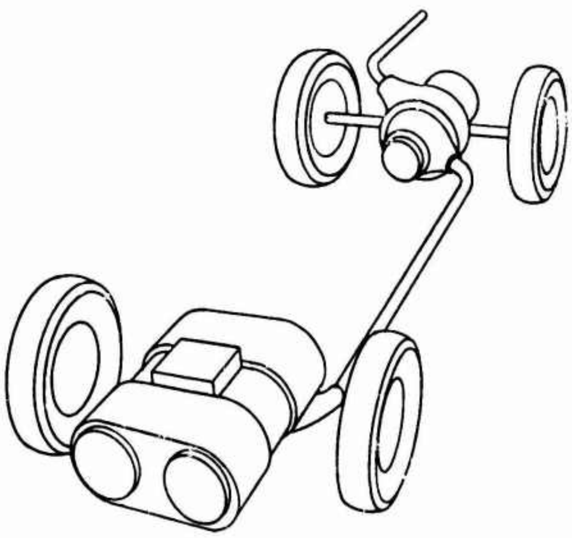


Fig. I-250—Schematic Diagram of Remote Gasifier and Turbine Unit in Automotive Vehicle

The free-piston turbine engine compares favorably in many respects with other types of power sources.

The output characteristics of the free-piston turbine engine are ideally suited to the power requirements of automotive vehicles. Figure I-255 compares the torque characteristics of the free-piston turbine engine and the piston engine. The curves indicate that the free-piston turbine engine delivers increasing torque down to turbine stall speed. This is in contrast to the piston engine, which delivers its maximum torque at 40 to 60 percent of its maximum speed and power. The ability to provide good torque at stall and near-stall speeds eliminates the need for a clutch or torque converter. This reduces the size and weight of the transmission unit required for a vehicle.

The thermal efficiency of the free-piston engine is higher than that of the more conventional power sources, such as the spark-ignition engine, compression-ignition engine, and the gas-turbine engine. Table I-26 illustrates a comparison of these power sources with respect to air-cycle thermal efficiency and actual thermal efficiency. Figure I-256 compares the thermal efficiencies of various power sources. The high efficiency of free-piston engines is largely

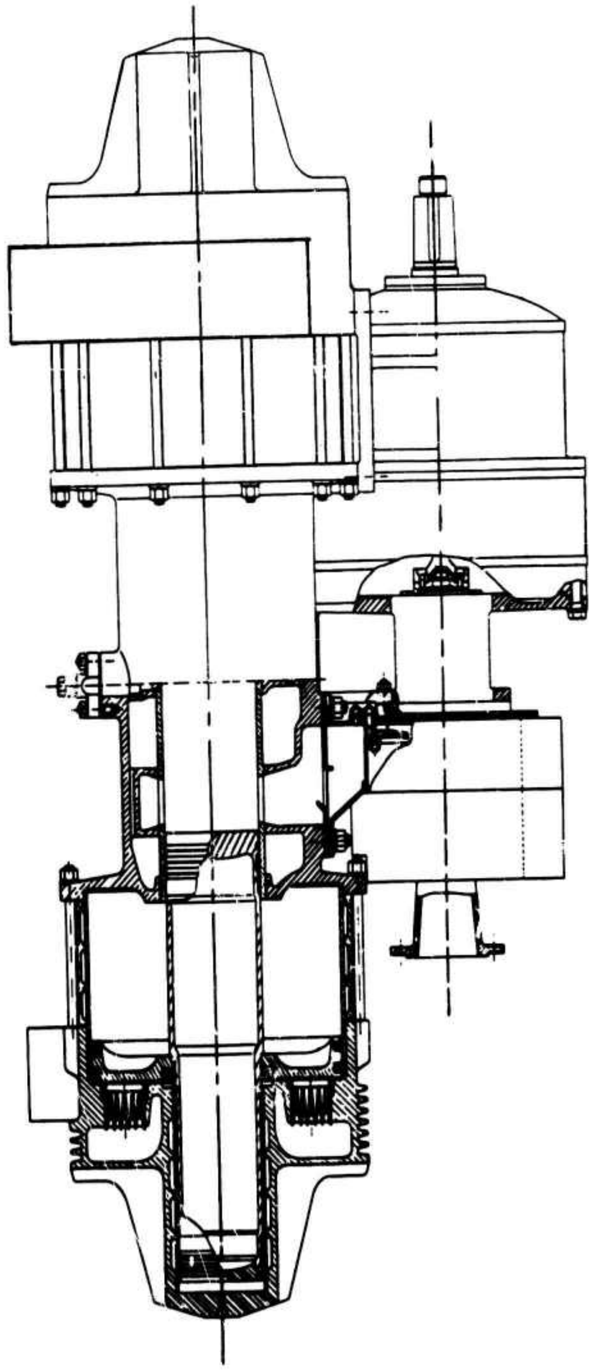


Fig. I-251—General Arrangement of Proposed 150-shp Free-Piston Turbine Engine with Transmission

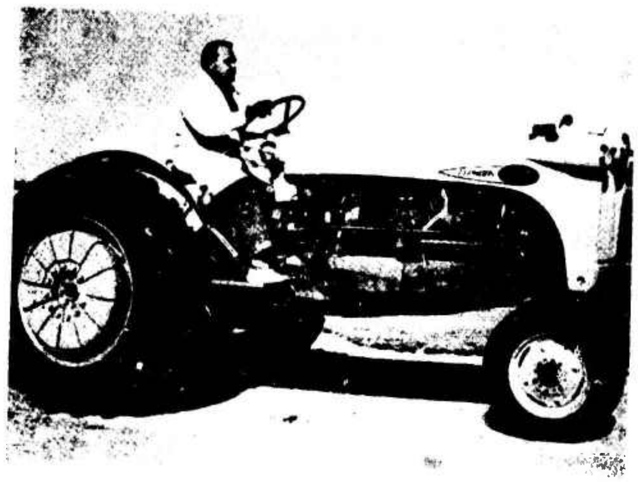


Fig. I-252—Free-Piston Turbine Engine Installed in a Farm Tractor

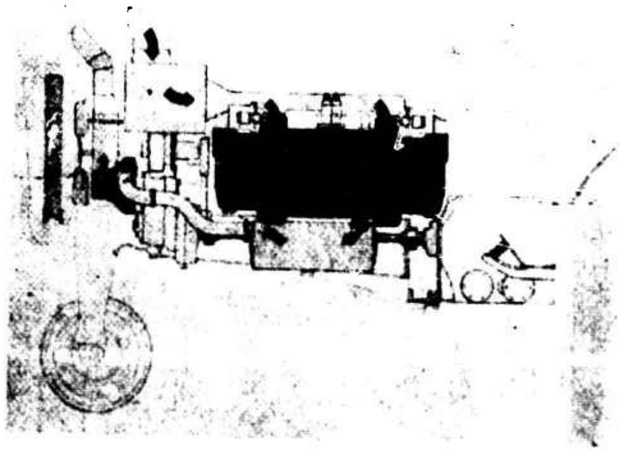


Fig. I-253—General Arrangement of Gasifier, Turbine, Gearbox, and Accessories in Farm Tractor

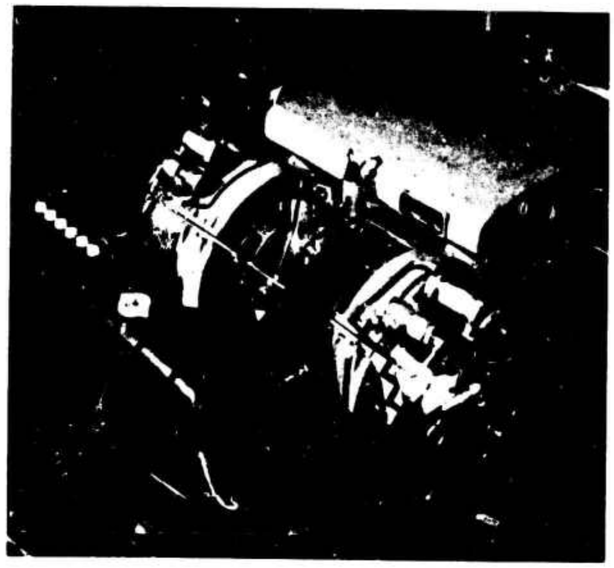


Fig. I-254—Free-Piston Turbine Engine Installed in Passenger Vehicle

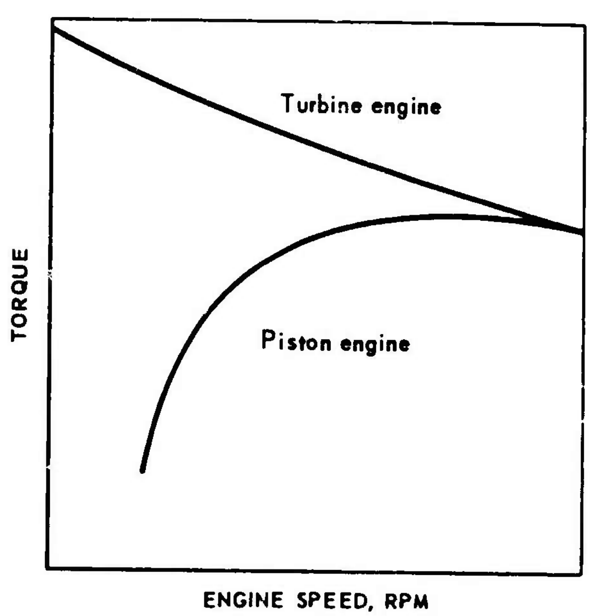


Fig. I-255—Comparison of Torque Characteristics of Turbine Engine and Piston Engine

TABLE I-26

Comparison of Efficiencies of Several Power Sources

(Present technology)

Characteristic	Type of power source			
	Gas turbine (regenerated)	Spark-ignition engine (gasoline)	Compression-ignition engine (diesel)	Free-piston engine (diesel)
Compression ratio (r_v)	7:1	10:1	22:1	40:1
Air-cycle thermal efficiency,[a] %	54	60	71	77
Actual thermal efficiency,[b] %	33.7	27.5	35.4	36.4

[a]Theoretical air-cycle thermal efficiency based on $\eta = 1 - \left(\frac{1}{r_v}\right)^{\gamma - 1} = 1 - \left(\frac{1}{r_v}\right)^{0.405}$

where $\gamma = \frac{\text{heat supplied} - \text{heat rejected}}{\text{heat applied}} = 1.405$

[b]Based on $\eta = \frac{\text{horsepower (ft-lb/hr)} \times 100}{f_{SFC}\ \text{(lb/hp-hr)} \times h\ \text{(Btu/lb)} \times \text{ft-lb/Btu}} = \frac{33{,}000 \times 60 \times 100}{f \times h \times 778.6}$ percent

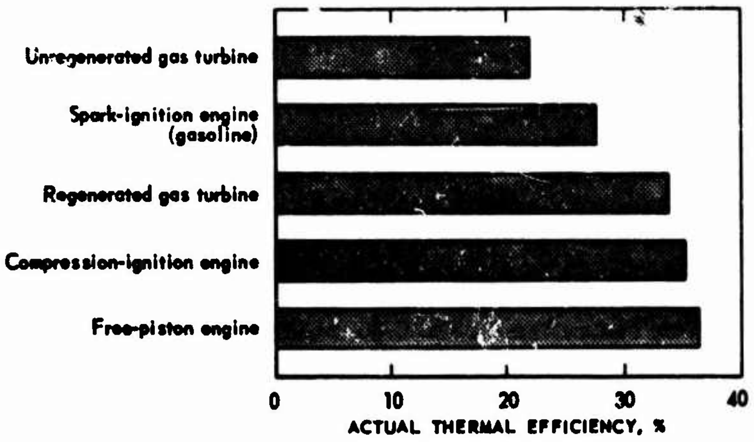


Fig. I-256—Thermal Efficiencies of Several Power Sources

Based on present units in 300- to 600-hp class.

due to the high adiabatic efficiency yielded by their high compression ratio and to the large quantities of excess air for combustion and for scavenging provided to the highly supercharged power cylinder.[2]

The fuel-consumption characteristics of free-piston engines vary with the type of gasifier design. Figure I-257 compares the full-load fuel consumption

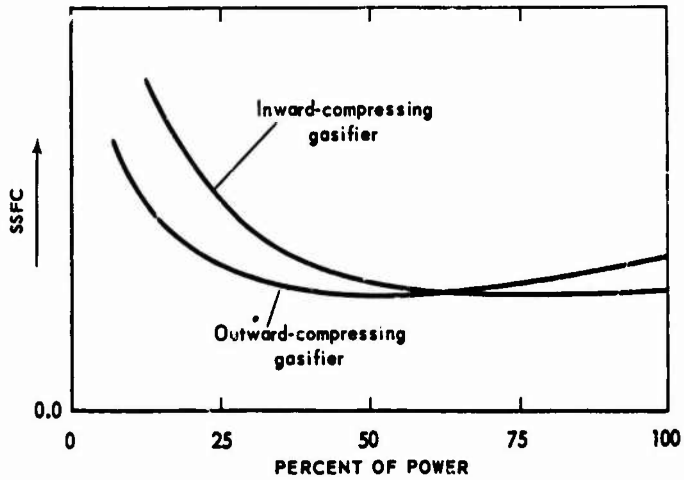


Fig. I-257—Comparison of Full-Load Fuel Rates of Typical Inward- and Outward-Compressing Free-Piston Engines

of the inward- and outward-compressing gasifiers. The curves show that from 60 to 100 percent power the inward-compressing gasifier has a lower fuel rate than the outward-compressing type and that the outward-compressing type has a lower fuel rate in the lower power ranges. The fuel consumption of small (100- to 400-hp) free-piston engines has generally been greater than that of compression-ignition engines. The automotive engines, such as the Ford 519 and the GMC Hyprex units, were reported to have demonstrated SSFC of 0.42 to 0.45 lb/hp-hr, which would correspond to approximately 0.48 to 0.51 lb/shp-hr, which is comparable to the fuel consumption of a spark-ignition gasoline engine. The automotive companies believe that with further development the fuel consumption can be reduced to 0.35 to 0.36 lb/ghp-hr, and with the incorporation of more advanced turbines an SSFC of approximately 0.42 lb/shp-hr can be attained. Figure I-258 illustrates the performance characteristics of the Muntz inward-compressing free-piston gasifier, which developed 450 gas hp.[3] The fuel-consumption curve indicates a minimum part-throttle SSFC of 0.328 lb/ghp-hr and a full-throttle SSFC of 0.345 lb/ghp-hr. Assuming a turbine efficiency of 80 percent, a minimum part-throttle SSFC of 0.41 lb/shp-hr and a maximum output of 360 shp could be achieved. The fuel-consumption rate of this engine is generally comparable to that of a compression-ignition engine of the same power rating. Figure I-259 illustrates the fuel-consumption characteristics of the Free Piston Development Co., Ltd., Series 5000 gasifier.

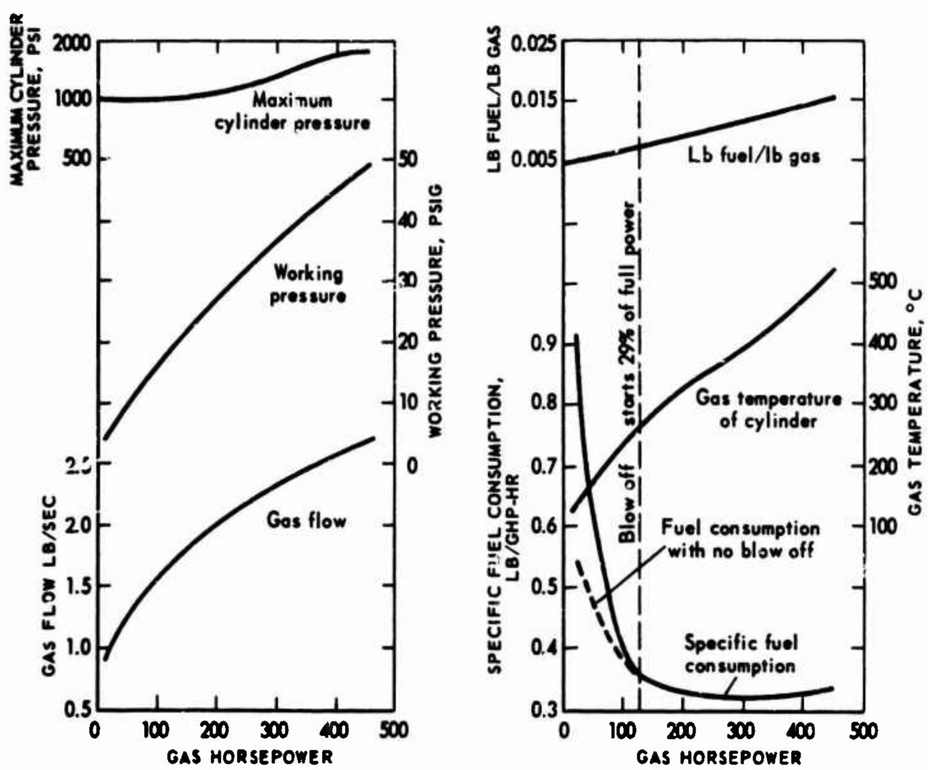


Fig. I-258—Performance Characteristics of Muntz 450-ghp Free-Piston Gasifier

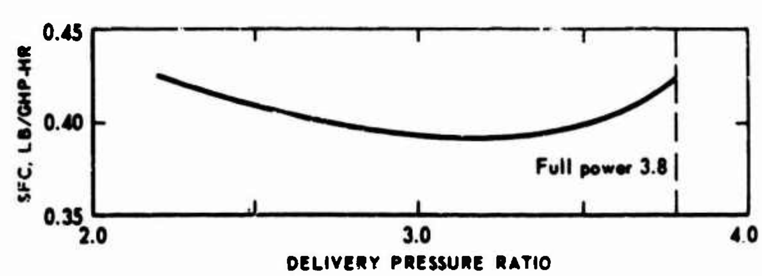


Fig. I-259—Fuel Consumption of Free Piston Development Co. Series 5000 60-ghp Free-Piston Gasifier

The curve indicates a best-point SSFC of 0.385 lb/ghp-hr and a full-throttle SSFC of 0.430 lb/ghp-hr. If this engine were coupled to a gas turbine of 85 percent efficiency, the best fuel-consumption rate would be approximately 0.45 lb/shp-hr. This performance lies between that of the compression-ignition engine and the spark-ignition gasoline engine. The developers of the Series 5000 gasifier assert that this fuel consumption could be lowered considerably.[3] Table I-27 illustrates the present thermodynamic performance of the Series 5000 gasifier. The gas-horsepower figures in this table are for a single-cylinder unit. The horsepower can be doubled or tripled by "twinning" or "tripling"

TABLE I-27

Thermodynamic Performance of Series 5000M Free-Piston Gasifier

Item	Value	
	Present	Developed
Delivery pressure ratio P_{del}/P_a	3.8	5
Delivery temperature T_{del} °R	1356	1435
Frequency, cycles/min	2015	2330
Mean piston speed, ft/min	1575	1850
Mass flow M, lb/min	26.0	33.8
Trapped air/fuel ratio R_T	31.8[a]	35
Gas thermal efficiency η_g	0.342	0.476
Gas specific consumption, lb/ghp-hr	0.409	0.294
Gas horsepower, ghp	64.6	100.0
Diesel indicated efficiency, %	0.40[a]	0.46
Mechanical efficiency, %	0.74[a]	0.79
Compressor efficiency, %	0.87[a]	0.90
Pressure loss factor F	1.22[a]	1.20
Fractional cylinder heat loss	0.33	0.15

[a]Estimated.

the gasifiers into a common unit. Table I-28 lists the specifications of the present-development free-piston engine and probable engines that could be available if their development were pursued. The data in this table are based on investigations of advanced-development engines by the Free Piston Development Company, Ltd. An SSFC of 0.346 lb/shp-hr for the Series 5000M 85-shp gasifier is based on present technology. Afterburning can be incorporated into this engine and would increase the output to 109 shp, but would also increase fuel consumption to 0.393 lb/shp-hr. The technique of using fuel in an afterburner is not as efficient as adding the fuel to the diesel section of the engine. Estimated turbine and reduction-gearbox weights listed in Table I-28 appear to be high. The operating characteristics of these components are similar to those of a conventional gas-turbine engine, and similar components may be used. Based on this reasoning, it appears that the weight of these components in the table may be reduced by approximately 50 percent, which would reduce the weight of the complete engine by 15 to 20 percent.

TABLE I-28

Specifications of Free-Piston Turbine Engines

Item	Present-development small gasifier	Engines that could be available, with development									
		Small				Large					
		Gasifier, lightweight version		Gasifier, lightweight, with after-burner (400°F rise)		Gasifier		Hot gasifier		Hot gasifier with after-burner (400°F rise)	
	Number of cylinders										
	1[a]	1[a]	2[b]	1[a]	2[b]	1[a]	2[b]	1[a]	2[b]	1[a]	2[b]
Diesel bore, in.	3.5	3.5		3.5		4.5		4.5		4.5	
Compressor bore, in.	9.5	9.5		9.5		12.2		12.2		12.2	
Gas horsepower	100	100	200	128	256	200	400	239	478	306	612
Shaft horsepower[c]	85	85	170	109	218	170	340	203	406	260	520
SSFC, lb/ghp-hr	0.294	0.294		0.334		0.294		0.246		0.280	
SSFC, lb/ehp-hr (turbine at 85% eff.)	0.346	0.346		0.393		0.346		0.289		0.329	
Gasifier weight, lb	410	268	270	473	480	389	685	389	685	405	720
Turbine weight, lb	150	150	210	150	220	100	100	100	110	100	120
Reduction gearbox weight, lb	150	150	210	150	220	200	200	200	220	200	240
Complete engine weight, lb	560	418	680	423	700	689	985	689	1015	705	1080
Specific engine weight, lb/shp	6.6	4.9	4.0	3.9	32.2	4.1	2.9	3.4	2.5	2.7	2.1
Engine volume (complete; approximate), ft^3	6.3	6.3	11.5	6.3	11.7	11.7	20.1	11.7	20.1	11.7	20.1
Specific power output, hp/ft^3	13.5	13.5	14.8	17.3	18.6	14.5	16.9	17.3	20.2	22.2	25.9

[a]Single-cylinder values based on Free Piston Development Co., Ltd., data.

[b]Twin-cylinder values are RAC estimates and are based on single-cylinder data.

[c]Actual values of shaft horsepower would be some 10% lower, due to the reduced swallowing capacity of inward-radial-flow turbines at speed, compared with stall. The given gas horsepower corresponds to maximum mass flow, i.e., at turbine stall.

A so-called "hot" engine, shown in Fig. I-260, has been proposed by Free Piston Development Co., Ltd.[4] Cooling of the diesel and compressor sections of this engine has been intentionally suppressed by adapting a labyrinth seal in place of conventional piston rings, and incorporating vacuum pockets in both the fixed and reciprocating components. Incorporation of vacuum spaces

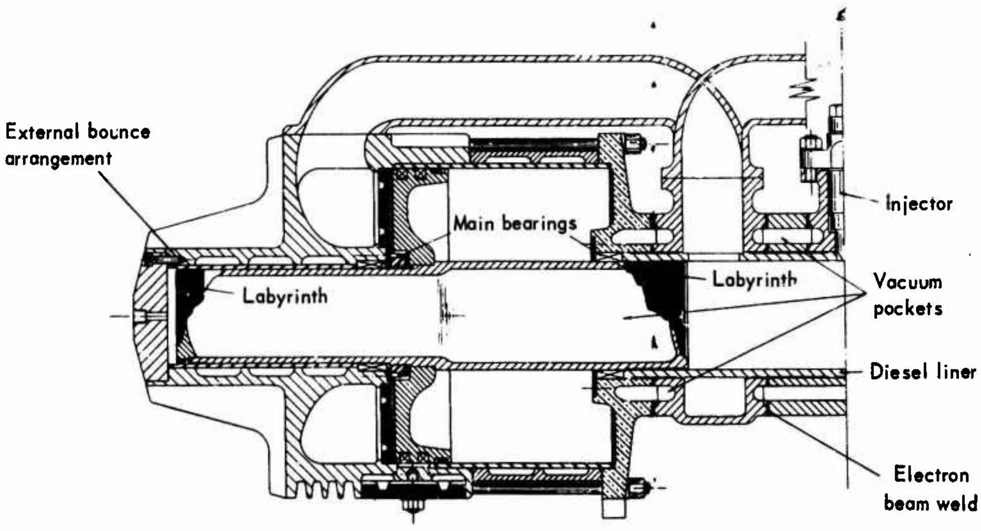


Fig. I-260—Cross Section of Proposed Hot Gasifier Utilizing Labyrinth Sealing of Pistons

in the diesel pistons and cylinder liners largely suppresses the flow of heat into the surroundings. This construction is possible since side loads on the pistons are virtually eliminated. This concept requires the use of high-strength high-temperature metal alloys. The uncooled gas-generator cycle, if developed, offers improvements over the conventional gas-generator cycle and the turbocharged and compound cycles. Figure I-261 illustrates the potential increase in the output of the hot gasifier as compared to those of other engines. A comparison in overall efficiency of the hot engine with other engines is illustrated in Fig. I-262, which indicates an overall efficiency of 46.8 percent for the hot-gas generator.[4] The output and efficiency increase appears only at very high boost pressures. Investigations show that the hot-gasifier engine should achieve better fuel consumption than conventional large gasifiers. The SSFC of the hot gasifier is projected to be 0.289 lb/shp-hr, as compared with 0.346 lb/shp-hr for the conventional gasifier. Further increases in power output can be obtained in the hot gasifier by afterburning. However, the SSFC will then increase to approximately 0.329 lb/shp-hr, since fuel is not used as efficiently in an afterburner as it is when added to the diesel cycle.

Free-piston engines compare favorably in size and weight with conventional compression-ignition engines. Free-piston engines built to date weigh

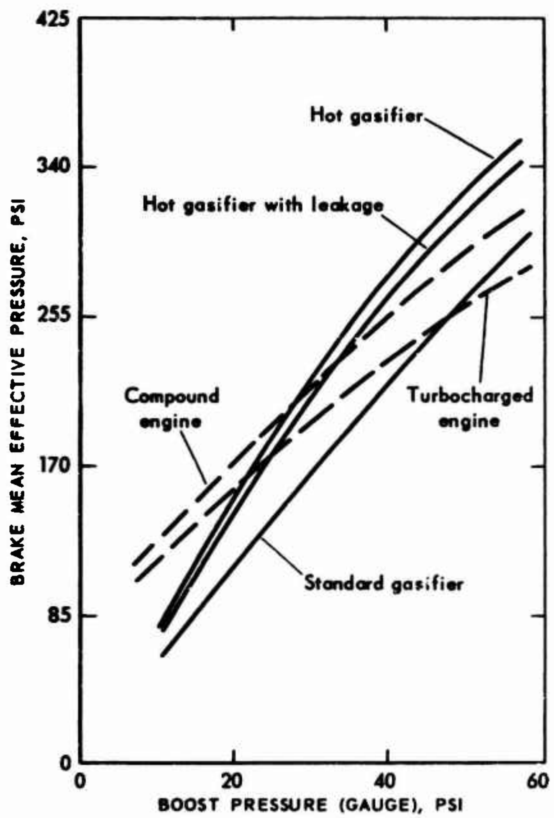


Fig. I-261—Comparison of Performance of Hot-Gas-Generator Cycle with Those of Other Engine Cycles

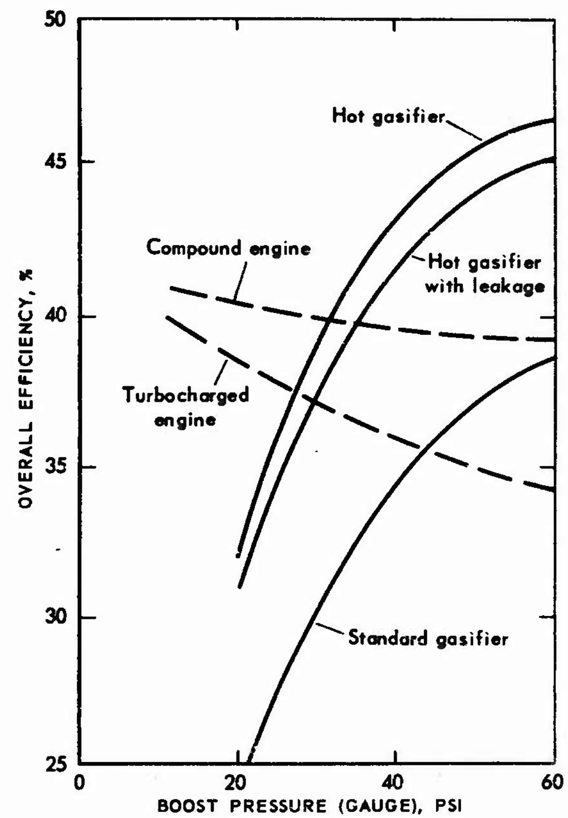


Fig. I-262—Comporison of Efficiency of Hot-Gas-Generotor Cycle with Those of Other Engine Cycles

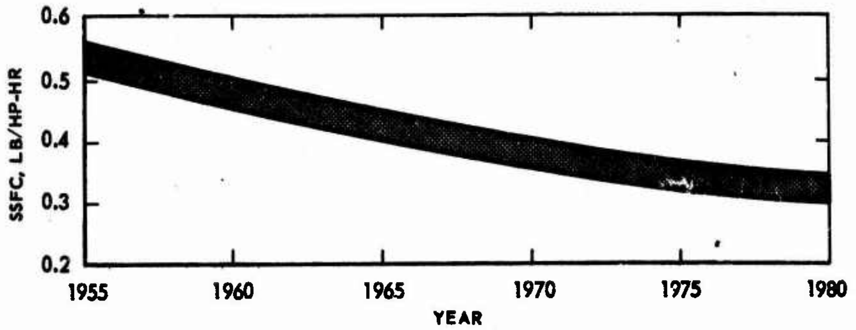


Fig. I-263—Trend Forecast of Fuel Consumption of Free-Piston Turbine Engines

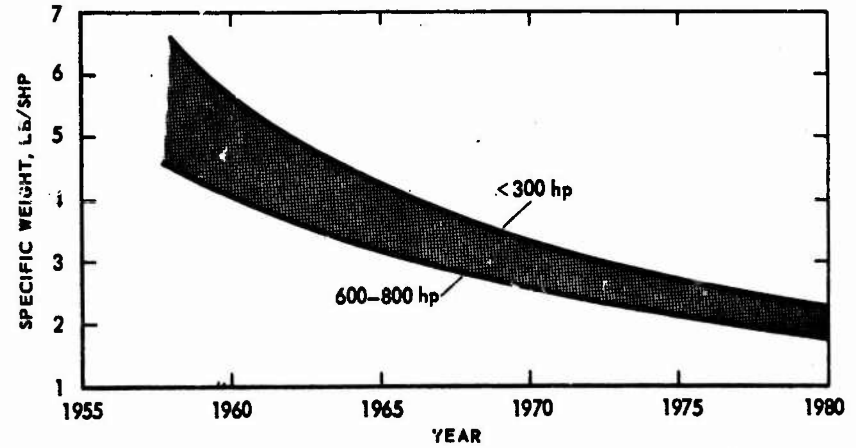


Fig. I-264—Trend Forecast of Specific Weight of Free-Piston Turbine Engines

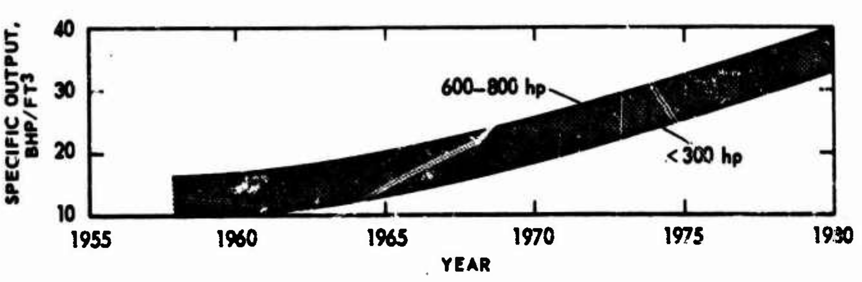


Fig. I-265—Trend Forecast of Specific Output of Free-Piston Turbine Engines from 4.5 to 6.5 lb/hp and have a specific output of 11 to 14 hp/ft^3. This is comparable to an average conventional compression-ignition engine with specific outputs of 5 lb/hp and 10 hp/ft^3. Both types of units compared are constructed of iron. If lightweight materials were used, the weight figures would decrease by approximately 20 to 22 percent. Additional development could further increase the power output of free-piston turbine engines. These improvements could be accomplished by

(a) Improving the gas cycle
(b) Increasing gasifier working pressures
(c) Increasing working temperatures
(d) Increasing turbine efficiency
(e) Using lighter materials

CONCLUSIONS

The free-piston turbine engine concept is feasible and practical. Its main advantage is its multifuel capability and low fuel consumption. Free-piston engines could have application in certain tactical vehicles where low fuel consumption is of greater importance than the size and weight of the power plant. Like free-spool gas-turbine engines, free-piston turbine engines have a characteristic that is similar to that of a torque converter. Further development of the free-piston turbine engine is required before it can be accepted as a vehicle power source. It appears that present deficiencies can be overcome by developing a better starting system, greater control, and more reliable reed valves, and by providing for a suitable accessory drive.

Commercial engine manufacturers show little or no interest in developing free-piston engines since they are more suited for military application. The military would therefore have to sponsor any development programs for free-piston engines if they are to be available for future tactical vehicles by 1980.

Free-piston turbine engines, with continued development, have the potential to:

(a) Reduce fuel consumption
(b) Attain a multifuel capability
(c) Increase power output at reduced weight
(d) Increase power output for engine size
(e) Achieve good reliability
(f) Reduce maintenance requirements

The trend-forecast charts shown in Figs. I-263 to I-265 illustrate the estimated level of technological achievement of the free-piston turbine engine through 1980.

It is concluded that Government support of a research and development program for a free-piston turbine engine is warranted.

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Chapter 16

KGG-CYCLE FREE-PISTON ENGINES

INTRODUCTION

The KGG or Kuhns gas-generator cycle is a recent concept of a novel gas-cycle and gasifier unit that can be used in conjunction with a turbine to supply shaft power, much the same as a conventional gas-turbine engine. This unit is at present in the research stage, and no hardware has been produced. Patent rights to the KGG-cycle gas generator are assigned to the Marquardt Corporation.

The free-piston KGG is comprised of a stationary casing and two oppositely reciprocating piston-shell assemblies. The operational concept is similar to that of a more conventional free-piston gasifier. Each of these shell assemblies is in turn comprised of an outer and inner shell assembly. The outer shell slides along the central tube on dry-film-lubricated bearings and within the exhaust gas seals of the casing. The inner shell also slides on dry-film-lubricated bearings along the central tube and forms two symmetrically opposed sets of compression and expansion chambers. These chambers are sealed by dry-film-lubricated piston rings. A toroidal combustion chamber located at either end of the inner shell is intermediate in each set of chambers to create a double-acting engine. Automatic reed valves govern the flows through this engine. The combustion-chamber valves are closed for one complete piston stroke to secure efficient combustion under nearly constant-volume conditions without external disturbances.

A starting cell containing a spark plug is used to initiate combustion during the first combustion cycle only. Combustion occurs by autoignition after the first cycle.

The KGG has inherent perfect dynamic balance, which enables the unit to operate smoothly and without vibration. The gasifier can be coupled to an axial or radial-inflow turbine similar to the more conventional free-piston gasifier. The turbine unit may be mounted integrally with the gasifier unit, as in Fig. I-266, or remotely located, as in Fig. I-267. Flexibility in location of the power unit (turbine) provides greater flexibility for vehicle design. The gasifier section may be designed as a single unit or as two or more smaller units coupled together for greater compactness. A speed-reduction gearbox is required to reduce the output-shaft speed (which is of the same order as gas-turbine engine speeds) for vehicular application. Accessories may also be driven from this speed-reduction unit. Operation of the engine turbine at or near stall

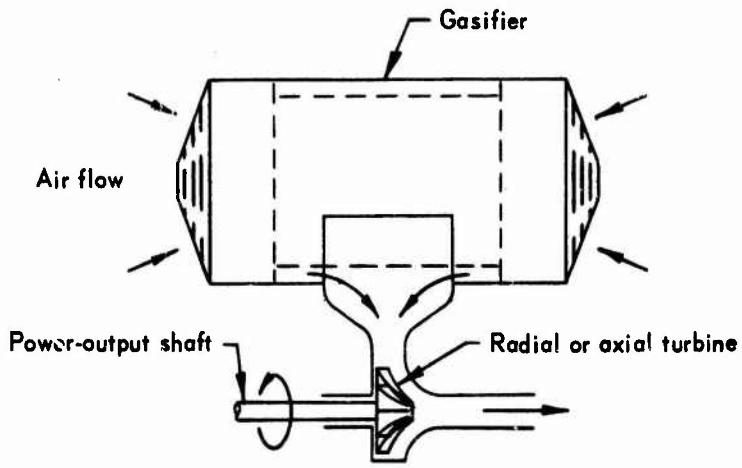

Fig. I-266.—KGG Engine with Integral Turbine

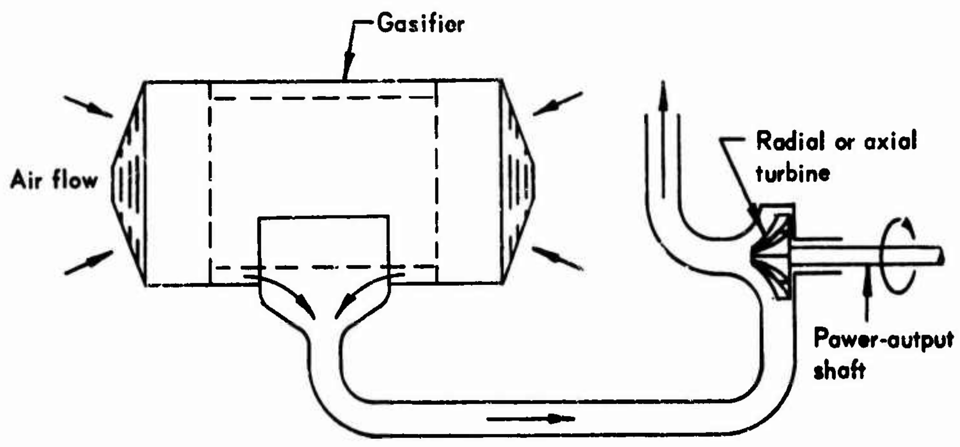

Fig. I-267—KGG Engine with Remote Turbine speeds would require a small gas-driven turbine or electrically driven accessory gearbox for most vehicular applications.

DISCUSSION

The KGG cycle is considered intermediate between that theoretically obtainable by pulsating-flow reciprocating-piston engines with constant-volume combustion and that by steady-flow gas-turbine engines with constant-pressure combustion. Figure I-268 illustrates a comparison of the KGG, Otto, and Brayton cycles. The KGG cycle appears to gain the advantage of thermal compression in a closed volume while avoiding the severe pressure rise of constant-volume combustion. It also has the advantage of decreased sensitivity to component losses while avoiding the high wall temperatures inherent in constant-pressure combustion.

The developer claims that the KGG cycle has a higher thermal efficiency than the gas-turbine (Brayton-cycle) gas generator when both have practical levels of component losses. Valve-pressure losses with the KGG-cycle gasifier will decrease with increasing compressor pressure ratio, whereas in the

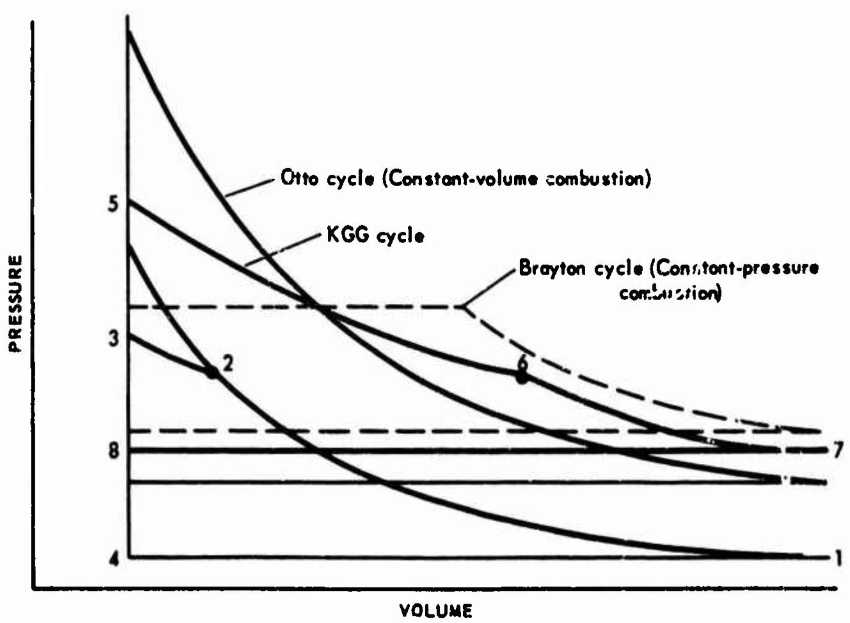

Fig. I-268—Comparison of KGG, Otto, and Brayton Cycles

gas-turbine gas generator the blade aerodynamic losses increase with increasing compressor pressure ratio. Considerable power-generating capability exists without initial compression, which enables the gasifier unit to be self-started without high-speed cranking. The thermal efficiency of the KGG and the gas-turbine gas generator are compared in Fig. I-269. The curves show that the KGG cycle has higher thermal efficiency than the gas-turbine gasifier over the full pressure-ratio range. Figure I-270 illustrates the horsepower per pound of air processed as a function of wall temperature for the KGG-cycle and Brayton-cycle gasifiers. The curves show that the KGG-cycle gasifier has a much higher power output at a given wall temperature than the Brayton-cycle gas generator.

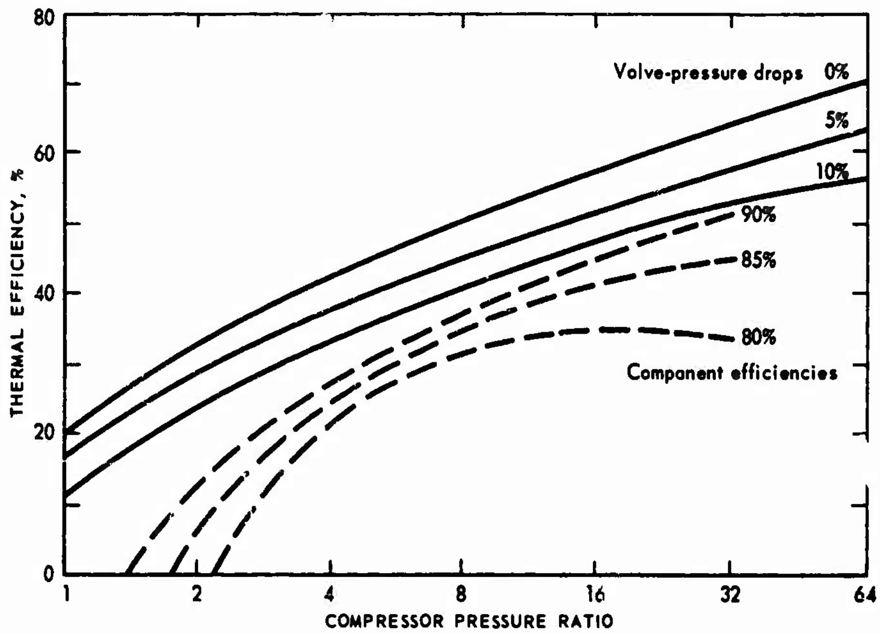

Fig. I-269—Thermal-Efficiency Comparison of KGG with Gas-Turbine Gas Generator

The single-stage piston-expansion-cooling of hot gases in the KGG gasifier results in much lower uncooled wall temperatures than in a gas turbine. Cooling the compact KGG parts, which are subjected to intermittent flows at relatively low velocities (compared with the large blade areas continuously subjected to extremely high velocities in a gas turbine), should be relatively easy to accomplish.

Figure I-271 illustrates the effect of combining the characteristics shown in Figs. I-269 and I-270 and compares the thermal efficiencies of a KGG with a gas-turbine gas generator. The curves indicate that the KGG has a thermal efficiency approximately 50 percent higher than that of the gas-turbine gas generator at midrange pressure ratios.

Very high component efficiencies are claimed for the KGG-cycle gasifier by the engine developer. The internal performance of the piston components is in theory highly efficient. The toroidal vortex combustor should be able to achieve nearly complete burning of fuel. The combustion process, without external or internal flow disturbance, proceeds for approximately 3 msec, which is comparable with both the diesel and the gas turbine. Studies conducted by the Marquardt Corporation indicate that, owing to relatively low flow velocities and straight-line flow paths between the valves, the burning losses are negligible

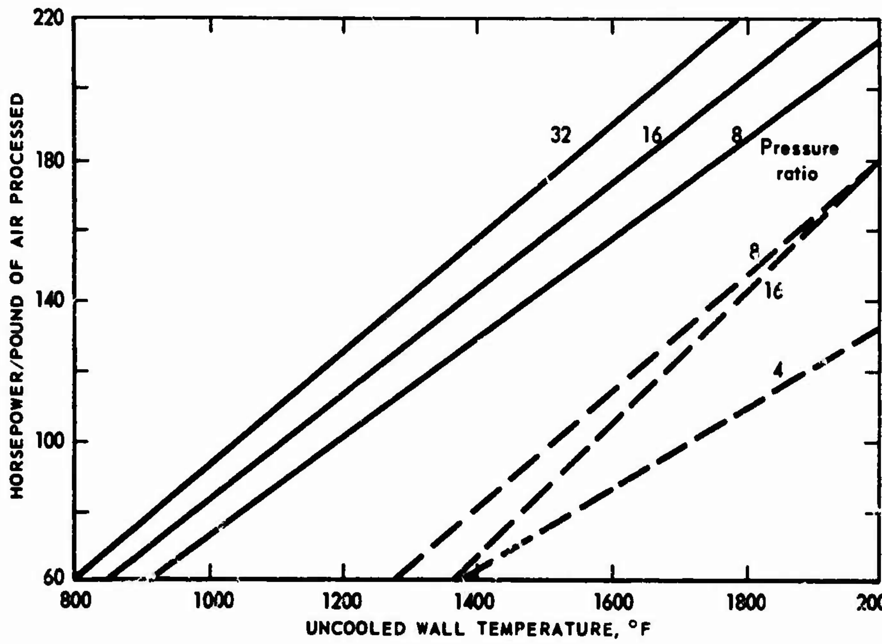

Fig. I-270—Power Comparison of KGG with Gas-Turbine Gas Generator

——— Kuhns gas generator   — — Gas-turbine gas generator

in the KGG gasifier. The sudden-expansion losses for the KGG-cycle gasifier concept, as a function of piston speed and intake-valve/piston area ratio, are shown in Fig. I-272. The chart indicates that at extreme speeds valve areas less than half the piston area result in a pressure loss of approximately 5 percent. If this figure is correct, the KGG unit appears to yield higher theoretical overall efficiencies than gas turbines. The large valve areas in the KGG unit are in sharp contrast to those of conventional reciprocating engines, which may have intake valves with an open area of approximately 10 to 15 percent of piston area. The Marquardt Corporation studies indicate that it is feasible to consider open valve areas as high as 80 percent of the piston area. However, this figure appears to be highly optimistic.

The KGG-cycle gasifier is of lightweight construction and is comprised of approximately 100 parts, including controls and automatic interlocks.

The developer estimates that a typical KGG-cycle gasifier in its final form would produce 1000 ghp from a unit several times lighter and more compact than a conventional axial-compressor-turbine gasifier. The developer's estimate of the weight and volume of this unit appears to be very optimistic, although it is for a very-high-performance aircraft-type unit. The unit would have a SSFC of less than 0.34 lb/ghp-hr. A turboshaft version of this unit with a turbine efficiency of 85 percent would produce 850 shp at a specific fuel consumption of approximately 0.35 to 0.36 lb/shp-hr. Table I-29 lists the esti-

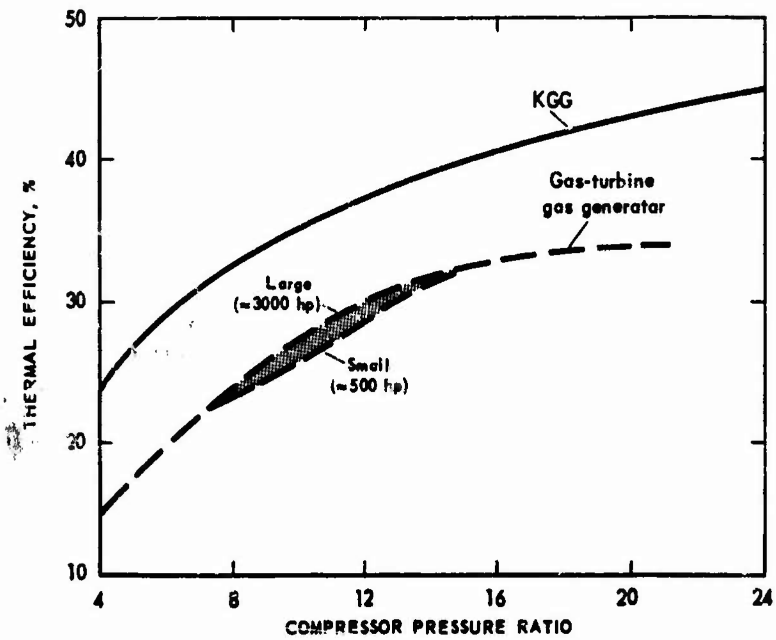

Fig. I-271—Practical-Efficiency Comparison of KGG with Gas-Turbine Gas Generator
Based on an uncooled wall temperature of 1600°F.

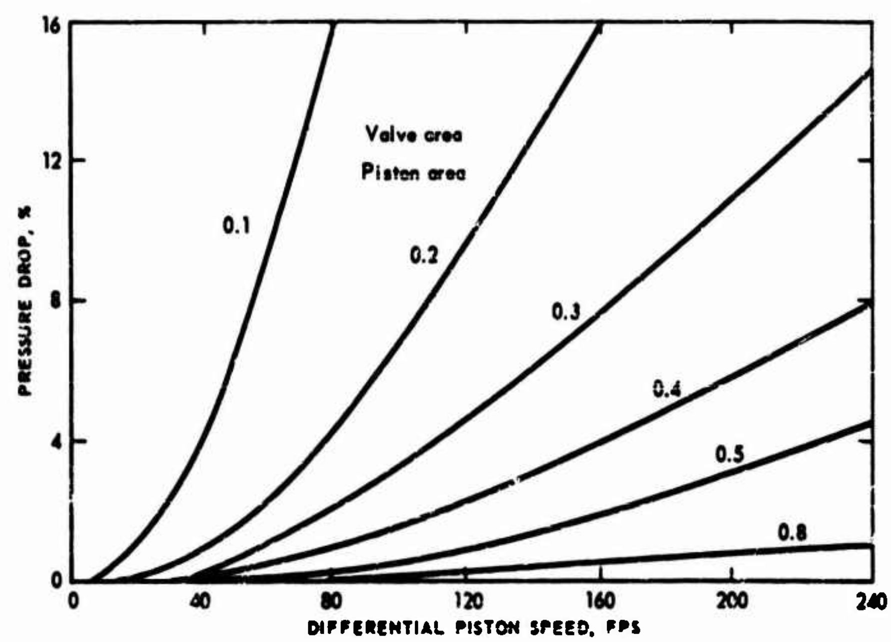

Fig. I-272—Valve-Pressure Drop as a Function of Differential Piston Speed for KGG-Cycle Gasifier

mated engine characteristics of realistic vehicular power plants in 250-, 500-, and 1000-shp sizes, which could be available in 8 to 10 years of concentrated development effort.

TABLE I-29

Estimated Characteristics of KGG Turbine Engines

Item	Shaft horsepower at 3500–4000 rpm		
	250	500	1000
Weight, lb (includes gasifier unit, power-turbine rotor, turbine case and support, reduction and drive gearbox, accessory-drive assembly, gas ducts, and all controls)	137	225	400
Specific weight, lb/shp	0.56	0.45	0.38
Volume, ft^3	4.2	7.2	12.5
Specific volume, hp/ft^3	60	70	80
SSFC at 100 percent power, lb/shp-hr	0.37–0.38	0.35–0.36	0.34–0.35
Minimum SSFC, lb/shp-hr	0.36–0.37	0.34–0.35	0.33–0.35

Most of the parts that make up the gas generator are circular and can be easily cast or readily formed from sheet metal. Their design and function permit wide tolerances. The turbine wheels and reduction gearboxes are similar to those presently used in gas-turbine engines and would require low development costs. In addition the use of a conventional starter system and complex lubrication systems could be avoided.

CONCLUSIONS

The KGG cycle appears to be feasible and workable. However, many design parameters must be considered and evaluated prior to construction of components for unit testing. Comprehensive analytical efforts are required in the following areas:

(a) Thorough cycle analysis
(b) Peak cycle temperatures
(c) Maximum wall temperatures
(d) Optimum pressure ratio
(e) Piston frequency
(f) Proper balance of stress levels, fatigue, and creep limits in lightweight metal parts

In addition to the above analytical studies, many areas of design would require major development effort. Some of these are:

(a) Inlet and charge valves
(b) Discharge valve
(c) Exhaust valve
(d) Seals
(e) Bearings and lubrication
(f) Materials selection

The future potential of the KGG turbine engine, with continued and unlimited development, could result in the following achievements:

(a) Reduced fuel consumption

(b) Multifuel capability
(c) Increased power output at reduced weight
(d) Increased power output for engine size
(e) Reduced maintenance
(f) Lower production cost than present engines

The forecast charts, shown in Figs. I-273 and I-274, illustrate the estimated level of technological achievement of the KGG turbine engine through 1980.

The development of the KGG turbine engine is considered to be a high-risk venture. There appears to be a lack of interest in developing this unit on the part of the US Air Force because of the risk involved and skepticism on achieving technological advancements beyond those already provided by the gas-turbine jet engine.

It is concluded that the US Army should not sponsor the development of the KGG turbine engine for tactical-vehicle application until a thorough cycle and mechanical analysis can be conducted or development of the engine is sponsored by the Government for use in aircraft. If these efforts are successful, the engine should be considered for use in tactical vehicles. If vehicular application is judged promising, the engine would then require additional development.

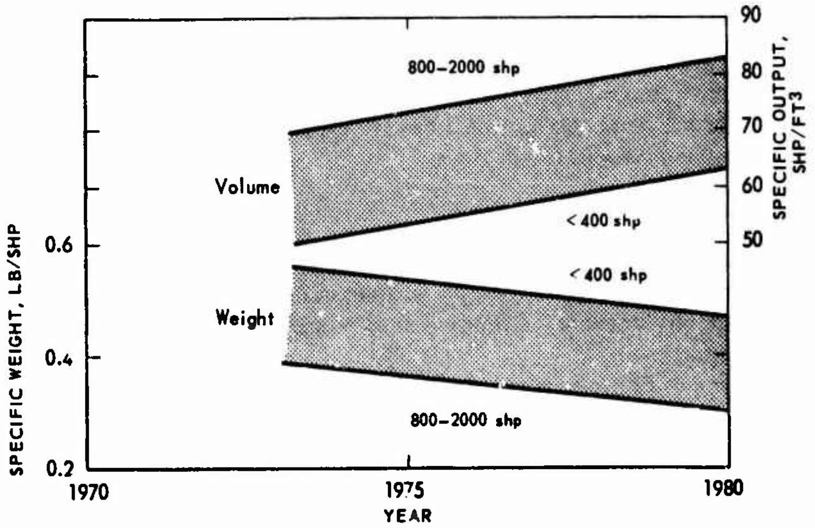

Fig. I-273—Trend Forecast of Specific Weight and Specific Output of KGG-Cycle-Gasifier Turbine Engine

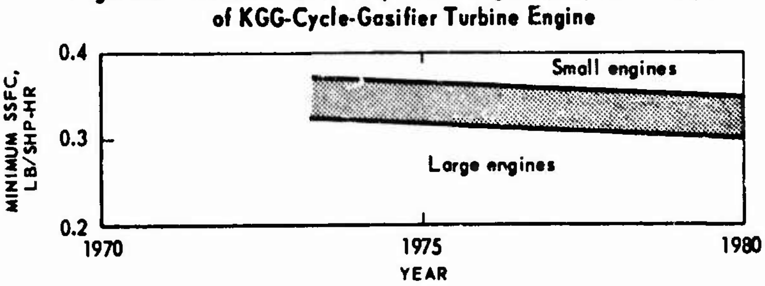

Fig. I-274—Trend Forecast of Fuel Consumption of KGG-Cycle-Gasifier Turbine Engine

PART II

Power-Conversion Devices

Chapter 17

INTRODUCTION TO PART II

For the purpose of this study, the phrase "power-conversion devices" includes all those components required for a system to transmit power from an energy-conversion device to the driving wheels or sprockets of a vehicle.

Although the primary function of a power-conversion device is to transmit power, other basic functions must be provided for acceptable vehicle operation. The power-conversion device must be capable of

(a) Changing the output speed and torque of the energy-conversion device.
(b) Engaging or disengaging the power.
(c) Being readily controlled by the driver.

To accomplish these basic functions efficiently, power-conversion devices are designed with a number of operating-speed ranges to match the speed and torque characteristics of the energy-conversion device.

Steadily increasing vehicle performance requirements have continued to place higher demands on power-conversion devices, which have resulted in the development of many new concepts.

In recent years emphasis has been placed on higher vehicle speeds and acceleration for improved vehicle agility. In addition a concerted effort has been made by the US Army to improve the physical and performance specifications of tactical vehicles. Vehicles must be capable of swimming in inland waters, be transportable by air and exert low ground pressure in order to improve mobility. These requirements have resulted in the development of power-conversion devices that are more complex in design, more costly to produce and maintain, more efficient, smaller, and lighter than their predecessors. However, their size and weight in relation to their power output must be reduced further in order to meet the goals set for future tactical vehicles. Since space in tactical vehicles for power-conversion devices is limited, they must transmit greater power without an increase in weight or size, or, if emphasis is placed on reducing the vehicle's weight, they must transmit the same power with reduced weight and size.

Tracked vehicles require a feature that is not required by most wheeled vehicles. This feature is the ability to create a controlled speed differential between the tracks to steer the vehicle. The addition of this feature makes power-conversion devices for tracked vehicles larger, heavier, and more expensive to produce than comparable power-conversion devices for wheeled vehicles.

A definition has been recently established by military personnel to specify power-conversion devices for wheeled vehicles as "transmissions" and power-conversion devices for tracked vehicles as "power trains" when steering and braking functions are included.

The five basic types of power-conversion devices discussed in this report are

(a) Mechanical (b) Hydrokinetic (c) Hydromechanical	Chapter 18 for wheeled vehicles, Chapter 19 for tracked vehicles
(d) Hydrostatic drive	
(e) Electric drive	

The first three power-conversion devices are discussed separately for wheeled and tracked vehicles; the last two are discussed together since they are applicable to both wheeled and tracked vehicles.

Chapter 18

MECHANICAL, HYDROKINETIC, AND HYDROMECHANICAL POWER-CONVERSION DEVICES

Information on mechanical, hydrokinetic, and hydromechanical power-conversion devices discussed was obtained from written reports prepared by industry and Government agencies and interviews with personnel from each of these two groups. This information was evaluated in an effort to ascertain (a) the power-conversion devices that will satisfy future tactical vehicle requirements, (b) the types and sizes of conversion devices not available for future tactical vehicle requirements, and (c) present concepts that may fulfill some known future requirements if R&D efforts were implemented.

A visit was made to ATAC in Warren, Mich. Discussions with personnel at this agency centered around current and future developments of military conversion devices. Some detailed information and technical data were received. Of particular interest was a chart on various power-conversion programs for vehicles from ¼ to 50 tons GVW in the past or now being implemented. At RAC's request, ATAC authorized visits to several industries that have military contracts concerned with power-conversion devices under ATAC's cognizance.

A visit was made to Allison Division of GMC, Indianapolis, Ind., to discuss various hydrokinetic power-conversion-device programs. New power-conversion-device concepts related to military vehicle requirements were discussed.

Contact was made with personnel at Fuller Division of Eaton, Yale and Towne Corporation of Kalamazoo, Mich. Discussions were held on their mechanical conversion devices pertaining to several new concepts that show promise of reducing device weight while increasing the performance.

Contact was made with General Electric Company, Ballston Spa, N. Y., and American Brake Shoe Corporation, Columbus, Ohio, for data on their hydromechanical programs, and vehicle test results were discussed.

Contact was made with Curtiss-Wright Corporation of Woodridge, N. J. Discussions were held on toroidal transmission, commonly called a "traction drive." This transmission was tested by ATAC for possible military application. Various deficiencies of the traction drive and the limits affecting application were revealed by both ATAC and Curtiss-Wright Corporation personnel.

Personnel at Spicer Division of Dana Corporation were contacted on their development of military transmissions. Their development programs are primarily directed to improve component designs. Their success in such programs could improve reliability and overall efficiency.

WHEELED VEHICLES VS TRACKED VEHICLES

Power-conversion devices of wheeled and tracked vehicles have been analyzed separately, since tracked vehicles provide a device through which steering is accomplished. The steering unit can be either an integral part of the power-conversion device or a separate assembly. In either case it is considered part of the power-conversion device for a tracked vehicle. With respect to equally rated wheeled-vehicle units and tracked-vehicle units the tracked-vehicle unit is, in general, heavier, bulkier, more complex, and more costly. Since a comparison of wheeled and tracked vehicles cannot be made on an equal basis, they are discussed and referred to separately in Part II as "Transmissions for Wheeled Vehicles" and "Power Trains for Tracked Vehicles."

MECHANICAL TRANSMISSIONS FOR WHEELED VEHICLES

Mechanical Transmissions

Mechanical transmissions utilize only mechanical components to transmit and convert the power from the power plant in the form of speed and torque to their points of application, which are the driving wheels of the vehicle.

These are analyzed in the sequence of their development:

Mechanical Transmissions:

(a) Progressive sliding gear
(b) Selective sliding gear
(c) Constant mesh
(d) Synchromesh
(e) Traction drive
(f) Belt drive

The elements and mechanisms adapted in a mechanical transmission are based on known principles developed a long time ago. However, application of the transmission concept as such was not made until 1834, when a number of gear sets having different ratios were assembled in the first "gearbox" invented. This design achieved torque multiplication and speed changes for vehicular application.

The fundamental functions and principles of the elements used in this first design such as gears, shafts, bearings, clutches, etc., have remained unchanged during the evolution of transmission design. The history of the development, however, has been marked with a sequence of component and component relation discoveries and practical inventions that have made constant improvements in transmission capability. These transmissions have undergone continuous modification from the data of the first design, which was ~~quite~~ inefficient, bulky, and heavy. Units today are considered reasonably priced, efficient, compact, light in weight (on a comparative basis), and reliable for vehicular application.

Progressive Sliding-Gear Transmission

The progressive sliding-gear transmission was the first type of transmission used for both commercial and military vehicles. "Progressive sliding" describes a transmission principle. A number of gear sets are arranged so that gear shifting can be performed only in a progressive order. The mechanisms do not permit a shifting from the first set of gears to the third without first shifting through a second set of gears. The same shifting sequence applies, in the reverse, for down shifting. The design is quite simple and results in an efficient, trouble-free, and compact unit for the power that it is capable of transmitting.

The progressive sliding-gear transmission had disadvantages for vehicular usage. The sliding-gear transmission principle requires a straight-tooth gear design to allow engagement of rotating gears. Spur gears with this type of stub tooth are noisy in certain speed ranges, and gear clashing occurs during shifting operation. The shifting of gears is cumbersome and very inefficient from the standpoint of vehicle control. These disadvantages outweigh the desirable characteristics of this type of transmission. Therefore, progressive sliding-gear transmissions are generally not used for vehicles except for some installations in motorcycles and other similar small vehicles.

Selective Sliding-Gear Transmission

The selective sliding-gear transmission superseded the progressive sliding-gear transmission and was incorporated in most military wheeled vehicles during and shortly after WWII.

The principle differs from the progressive sliding gear inasmuch as any selection of gear ratios is possible without passing through intermediate gear ranges. This improvement over the progressive sliding-gear principle provided better unit control and increased vehicle mobility. No other changes were incorporated into this unit to increase its capabilities over that of the progressive sliding-gear type. The principal components remained the same, and the noise and gear clashing were not eliminated.

There are still some vehicles with selective sliding-gear transmissions in operation, but generally no new vehicles are being built with selective sliding-gear transmissions. New types of transmissions have replaced them.

Constant-Mesh Transmissions

The next step in transmission development was the constant-mesh transmission. This transmission superseded the selective sliding-gear transmission and was adopted for military application after WWII.

Constant-mesh transmissions differ from selective sliding-gear transmissions mainly in the method of gear shifting. Selective sliding-gear transmissions follow the principle of meshing gears through axial sliding of one gear to mate with its counterpart; constant-mesh transmission gears are continuously in mesh. This permits the designer to use helical gears to provide smoother and quieter operation. These gears are splined to the same shaft as one of the free-wheeling main gears. The clutch gears are moved axially during the shifting operation until the external teeth engage with the internal teeth of the free-wheeling main gear. This action ties the main gear through the clutch gear to the shaft that transmits the power to the output shaft.

The use of helical gears in the intermediate and high-speed ranges has eliminated some of the transmission noises. However, gear-teeth clashing was not eliminated, and gear shifting is still accompanied by the usual gear-clashing noise during the engagement between the external clutch gear and internal main-gear teeth.

The use of constant-mesh transmissions for military vehicles has been greatly reduced in recent years, and their installation is now limited to a relatively small number of vehicles in the heavier weight classes. New concepts have been developed, and most of the constant-mesh transmissions have been replaced with synchromesh transmissions.

Synchromesh Transmissions

This type of transmission represents the latest design concept for mechanical transmissions.

The synchromesh design is based on the constant-mesh transmission principle that keeps main transmission gears constantly engaged. However, it incorporates additional features that eliminate gear clashing between mating parts and provides a much smoother power flow throughout the entire power range. This is achieved through a combination of friction and positive engagement clutches. The friction clutch synchronizes the speeds of rotating members after which the positive engagement clutch engages the driving and driven gear with the transmission main shaft. All this is accomplished during declutching and the gear-selector shifting operation.

Ever since their first installation in military and commercial vehicles, synchromesh transmissions have proved superior in operating characteristics over other previously mentioned transmissions, which accounts for their usage in almost all weight classes of military wheeled vehicles today.

Constant development of this transmission has resulted in its widespread use, and future development and improvements will be continued by industry. The basic principle has remained the same. Future improvements may not be as significant as they have been in the past, but further development will be continued by industry since synchromesh transmissions will remain acceptable for vehicular applications in the foreseeable future.

Traction Drives

The design of a traction-drive automotive transmission was first conceived in 1943, but it was not until 1959 that a useful model was constructed and tested in a vehicle.

The principle of a traction drive relies on the surface friction between two contacting elements to transmit motion and force from one to the other. Several different configurations have been designed, of which the commonly known are the following types:

- (a) Dual ring
- (b) Spool
- (c) Internal spool
- (d) Planet roller
- (e) Toroidal

All these developments have been used for various commercial applications, but only one, the toroidal traction drive, showed sufficient promise to be supported by the military for installation in a military vehicle.

The main components of the toroidal traction drive are two disks and three rollers. One disk is the driver or input member, the other the driven or output member. The rollers are located between the disks and are free to rotate around their own individual axes. They are constantly in surface contact with both disks, thus providing an uninterrupted infinitely variable ratio between the input and output shafts. These speed ratios vary from 1:1 to 8:1. This can be obtained by changing the roller position in relation to the common drive center line of the disks. Positioning of the rollers is accomplished hydraulically and activated through hydraulic pressure signals from the control system.

The toroidal traction drive cannot be used for vehicles unless a method is provided to disengage the drive and a device provided to reverse the direction of the output speed. Furthermore, only half the overall ratio of 8:1 can be fully utilized unless a 4:1 reduction gearbox is provided.

The construction of a toroidal traction drive is compact in design, but the overall unit size combined with a gearbox is larger and heavier than other mechanical transmissions of equal rating.

The efficiency of this friction drive is rated between 86 and 91 percent, not counting the power losses from the hydraulic pump that provides lubrication for the drive components. Assuming an average efficiency of 88.5 percent for the toroidal traction drive and 96 percent for the gearbox, the combined unit efficiency is less than that of a four- or five-speed synchromesh mechanical transmission.

From the standpoint of durability and judging by the present state of the art, traction drives cannot compete with other mechanical power-conversion devices. However, this may not be due to a faulty concept, but rather due to the difficulty of controlling the rollers during ratio changes. Most of the failures occurred owing to the lack of position traction from surface friction. The slightest misalignment of one roller results in its being out of phase with the others. This means that two different ratios are being engaged at the same time between input and output disks. When this occurs, one or the other rollers begins to slip, and the buildup of excessive heat breaks down the oil film between roller and disks, and the material spalls.

Military efforts on toroidal drives have been under the cognizance of ATAC. An experimental model combined with a gearbox was installed in an M151 jeep and endurance tested during highway and cross-country operation. The test was terminated in 1962 after 16,675 miles owing to roller control failures resulting in gross slippage and wear. A follow-on contract was awarded to provide a toroidal drive with improved controls for an additional evaluation test of 20,000 miles. That test began in October 1963 and terminated in 1964.

A preliminary design contract for a toroidal traction-drive transmission in a 5-ton XM656 vehicle was initiated. But the program was terminated when no concept was derived to overcome the roller control deficiencies encountered during the two tests of the M151 jeep.

Despite the advantage of an infinitely variable speed capability, it cannot be foreseen that the toroidal drive would be an improvement over other existing mobility requirements for on- and off-highway vehicles could not be fully accomplished with mechanical transmissions, and most vehicles are currently incorporating hydrokinetic transmissions.

Various concepts of the hydrokinetic transmission have been developed through the years. These are discussed in the sequence of their development:

(a) "Hydramatic."*
(b) Torque converter.
(c) "Torqmatic."*
(d) TX Series.

"Hydramatic" Transmission

The "hydramatic" transmission is fully automatic and utilizes a fluid coupling between the energy-conversion drive and the mechanical gear set. Many commercial vehicles employ this design in both passenger cars and trucks. The military vehicles use this transmission in various types of 2½-ton trucks in the 100- to 150-hp range.

Fluid couplings are hydrodynamic drives that transmit torque and speed without torque multiplication. There is always a certain amount of slippage and efficiency loss between impeller and reactor. The amount depends on the torque and speed that are being transmitted. At low speed the efficiency loss is the greatest, while at high speed the efficiency loss may be as low as 3 percent. The changing of gears is accomplished automatically by changes in hydraulic pressure, activated by either throttle control or vehicle-speed-sensing devices. This principle is commonly used throughout the various automatic hydrokinetic transmissions. Most "hydramatic" transmissions used in the fleet of military trucks provide four forward and one reverse speed. Some of these truck transmissions, however, provide power in one direction only, and reversing is accomplished within the transfer-case assemblies.

"Hydramatic" transmissions are compact and their specific rating and volumetric efficiency is not much lower than a comparable mechanical transmission. The overall mechanical efficiency of the "hydramatic" transmission is less, but this has been an accepted tradeoff for better vehicle control.

Since the first installation of "hydramatic" transmissions in military trucks, advances have been made in other types of hydrokinetic power conversion devices. These are gradually replacing "hydramatic" transmissions in military trucks.

Torque-Converter Transmission

The torque-converter transmission was designed and developed shortly after the "hydramatic" transmission was developed. Both utilize the same principle for the mechanical portion of the system and vary only in the hydrodynamic elements used.

Torque-converter transmissions incorporate a torque converter that provides the same function as a fluid coupling but, in addition, has the ability to multiply torque.

*GMC trademark.

The torque converter increases the overall transmission torque range that could otherwise be accomplished by the addition of a mechanical gear set but at an appreciable increase in weight. The torque converter has better efficiency than the fluid coupling during the high-torque—low-speed-range operations but efficiency decreases considerably at high-speed—low-torque-range operations. To overcome the decreased efficiency most torque converters are designed with an automatic lockout at the higher speeds that provides for a direct drive by bypassing the converter.

All hydrokinetic transmissions, with the exception of the hydramatic, incorporate a torque converter.

The automotive industry is using torque-converter transmissions in all passenger cars and light trucks that are equipped with automatic transmissions. The military may purchase some of these commercial vehicles but not for tactical purposes. Since industry is constantly improving torque-converter transmissions, there is no need for additional development by the Government.

"Torqmatic" Transmissions

"Torqmatic" transmissions are the "truck version" of torque-converter transmissions. Both transmissions are based on the same principle, but "torqmatic" transmissions are used for heavy-duty vehicles. Their power rating is higher than that of torque-converter transmissions, and their construction is much sturdier for rugged on- and off-highway operation.

"Torqmatic" transmissions are available as either automatic or semiautomatic and in a combination of the two. The normal speed ratio extends over a range of 20:1 through three forward speed gear selections. This ratio is less for reverse since it has but one speed selection. Units with higher overall speed ratios and gear selections are commercially available for very heavy vehicles and special-purpose vehicles.

Automatic transmissions can adjust their torque and speed if sufficient time is available to sense the vehicle's requirements. However, they cannot anticipate in time to negotiate an obstacle. This characteristic may impede the mobility of the vehicle in the time it takes for the automatic unit to shift into the proper range. A vehicle can become immobilized if enough momentum is lost and sudden increased output torque causes a loss of traction and spins the wheels.

The operator of a vehicle with a semiautomatic type transmission can select the operating gear prior to negotiating abrupt changes in the terrain. Lesser terrain obstacles can be negotiated without ratio changes since the torque converter will increase its torque output with decreasing vehicle speeds. There is some skill involved in shifting semiautomatic "torqmatic" transmissions, but the operator can acquire this skill rapidly.

TX Series

The TX series transmission is a torque-converter transmission that is manufactured by the Allison Division of General Motors. At the start of WWII most military wheeled vehicles were equipped with conventional type mechanical transmission, and due to the urgent wartime needs large numbers were procured from various manufacturers by the military. However, WWII operational experience convinced the military that cross-country vehicles needed greater operating capabilities and that the incorporation of available transmissions could no longer satisfy vehicle requirements.

A stepped-up program began in 1943 to develop new power trains for tracked vehicles. Shortly thereafter the basic principles of these new power trains were adopted for wheeled-vehicle application and by 1947 five torque-converter transmissions were produced to transmit power ranging from 60 to 850 hp.

The TX series of transmissions are the product of continued development programs of torque-converter transmissions. Since 1951 a number of improvements have been accomplished, but the basic design principle has not changed.

The design includes a hydraulic torque converter connected to a planetary gear train and clutch arrangement through which neutral, reverse, and three forward-speed ranges can be obtained. In addition, a splitter gear and clutch are intermediately located between the converter and the main gear system which doubles the number of forward ranges, extending the total gear coverage to six forward speeds. Range selection is accomplished by manual operation of a selector valve which controls hydraulic flow to engage or disengage any one of the four range clutches. Most transmissions have a built-in retarder to facilitate hydrodynamic braking during downhill operation. All torque converters are designed with an automatic converter-lockup feature and in addition provide for a power takeoff to drive auxiliary equipment.

With the development of TX series transmissions certain improvements over the original torque-converter transmissions were accomplished. First, many parts were standardized and made interchangeable within the TX series family. Second, better overall mechanical efficiencies were obtained; and, third, the unit size and weight were reduced without sacrificing reliability. Fewer parts, simplicity, and the redesign of many parts has made it possible to reduce the overall cost of these transmissions.

The present application of the TX series transmissions for tactical vehicles ranges from 2- to 15-ton payload capacity. These units have greatly improved overall vehicle mobility, and additional improvements can be expected in the future.

HYDROMECHANICAL TRANSMISSIONS

The hydromechanical transmission is presently under development although a few prototypes have been built and tested in 2½-ton 6 x 6 trucks as early as 1962. The hydromechanical transmission incorporates a mechanical section, hydrostatic section, and controls for the system. Although various components in each section are not new in principle and individually have been applied in other types of transmission installations, their interrelated use in the hydromechanical transmission concepts is unique.

The mechanical section is comprised of an input shaft, a planetary gear set, and an output shaft. The hydrostatic section is comprised of two hydrostatic pumps, piping with several valves, and pump actuators. The hydrostatic section eliminates the need of a torque converter permitting a direct coupling from an energy-conversion device to the mechanical section. The power, at this point, is split and directed through the mechanical and hydrostatic portion of the transmission unit. This power is then reunited at the planetary gear set, and, by superimposing the power output from the hydrostatic portion on to the mechanical portion, controlled speed and torque are obtained at the output shaft. Since the power output in the hydrostatic section is infinitely variable, restricted only by its physical limitation of various hydrostatic components, a multitude of speed and torque ranges can be obtained to meet the vehicle's power requirements. The integrated transmission and engine-control system automatically adjusts the transmission power output to the power required by the vehicle and permits the engine to operate at its most economical power range.

Present torque-to-weight ratios and torque-to-volume ratios for the hydromechanical transmissions are for practical purposes equal to those of hydrokinetic transmissions. The mechanical efficiency of the hydromechanical transmission is slightly lower than other comparable hydrokinetic transmissions. However, the overall system efficiency of the hydromechanical transmission is better than that of the hydrokinetic transmission, and fuel savings between 5 and 7 percent have been achieved. The reliability of both the hydromechanical and hydrokinetic transmissions is at present considered equal. However, hydromechanical transmissions have approximately 25 percent fewer parts than hydrokinetic transmissions and therefore better reliability of future developed hydromechanical transmissions is predicted. For the same reason less maintenance requirement is foreseen. The vehicle's ease of operation, incorporating a hydromechanical transmission, or incorporating a hydrokinetic transmission, is very satisfactory and is rated equally for both.

Although vehicles incorporating initial hydromechanical transmissions did not perform gradeability tests satisfactorily due to their limited torque range, this deficiency was overcome by the addition of a planetary gear set. Future hydromechanical transmissions will provide adequate torque to meet gradeability requirements.

EVALUATION OF WHEELED-VEHICLE TRANSMISSIONS

After evaluating mechanical, hydrokinetic, and hydromechanical transmissions, it was determined that many earlier mechanical and hydrokinetic transmission models are no longer procured for military vehicles, but a few of these transmissions procured during WWII are still in the system. They are gradually being replaced by new and improved transmissions.

In the group of present mechanical transmissions, the synchromesh transmission has the most merit for use in tactical vehicles. The traction and belt-drive transmissions were tested, but they were determined to be unacceptable for most tactical vehicles. Future development of both transmissions should be limited to small power outputs and where low cost is of prime interest.

In the group of present hydrokinetic transmissions, two types are favored for use in tactical vehicles. One is the torque-converter transmission for small passenger-type vehicles, and the other is the TX series transmission for heavier vehicles.

The hydromechanical transmission has great promise for improving the operating characteristics of future tactical vehicles. Although these units are not currently used for wheeled vehicles, test reports of an installation in a 2½-ton 6 x 6 truck are favorable, and their adoption for wheeled vehicles can be foreseen in the near future.

New improvements for transmissions are being constantly proposed by industry as well as Government agencies. Many such improvements could contribute to the operating capabilities of tactical vehicles. Therefore sufficient funds should be provided to test and evaluate various transmissions in vehicles by the Government. If found to have potential, further R&D should be implemented by the Government if industry has no incentive to fund such a program.

Foreseen Improvements

Transmissions are rated by their maximum input torque, speed, and volumetric efficiency. The latter indicates its compactness in terms of size or weight to its input-torque capacity. Improvements in better materials, machining techniques, and design configuration can enhance volumetric efficiencies.

It is apparent that synchromesh transmissions will continue to be used in tactical vehicles in the foreseeable future although their application will decline and be restricted for use in smaller vehicles up to 120 hp. Continuing improvement in their construction and fabrication methods has greatly contributed to their reduction of weight, size, and cost. Additional improvements will become more difficult, and much will depend on metallurgical advances in the future. Production volume will continue to be large for commercial applications and industry will continue with further development for this type of transmission. Therefore military development work should continue but it should be limited to military versions of commercial units to meet specific requirements for tactical vehicles.

Industry is currently developing a countershaft-type transmission which will offer higher input ratings while reducing the weight and size of this mechanical transmission. Compactness is achieved by an arrangement of two countershafts that divide the input torque through two smaller gear sets. Clutch travel for gear engagement is small and equal for all speeds. This allows for shaft length to be shortened,and,since only half the torque is transmitted,shaft loads and deflection can be kept to a minimum. The number of gear sets through which torque and speed are transmitted greatly affects the efficiency of mechanical transmissions. Other contributing factors are improved bearings, lubrication, and tolerances. Considerable effort is being made to improve gear cutting for equal load distribution and reduction of surface wear.

Since lubricating oils break down at elevated temperatures, effort is directed to reduce heat build-up and improve lubricants to withstand elevated temperatures. Oil companies such as Chevron Research Corporation of Richmond, California, are developing lubricants to withstand temperatures up to 400°F. The trend to improve lubricants will have an impact on future transmission designs.

Torque-converter transmissions have excellent applications for many commercial and military vehicles. They are reliable and can be purchased at a reasonable cost, and spare parts are readily available. Industry has the incentive to continue their development since there is a large commercial market.

The results of Government funding would be small compared to what industry is now providing. Therefore, R&D effort by the Government should be limited to modification of commercially available units.

A new "torqmatic" transmission is currently being developed utilizing twin-turbine converters. One of the turbines provides for high torque, low speeds, and the other turbine provides for low torque and high speeds. This development promises to improve fuel consumption by 7 percent and incorporates fewer parts, which should improve its reliability. Thirty percent of the parts are interchangeable with other "torqmatic" transmissions and will aid in standardizing military equipment. Many of the transmission components can be removed without the removal of the entire unit from the vehicle. This should improve ease of maintenance. Improved casting techniques have made it possible to design lighter components, and new clutch materials have greatly resisted wear and should increase clutch life. Seals are being improved to resist wear, and improvements in metallurgy will increase the reliability and life of transmission gears.

The TX series transmissions are available for use in military vehicles ranging from 2½- to 25-ton capacity. They are capable of handling up to 700 lb/ft input torque and feature hydraulic retarders and parking brakes with good volumetric efficiencies. Physical space limitations in vehicles may not always permit the incorporation of a standard TX transmission, and modifications are being made to these units. Competitive procurement is difficult since only one manufacturer has developed the organization facilities and overcome the initial learning costs.

Since many tactical vehicles are produced in low densities, industry will not initiate design and development of new transmissions unless they can foresee a commercial application, or would be supported by the military.

The hydromechanical transmission is readily adaptable to various types of engines, requiring only minor adjustments in the control system. Although weight and size are approximately the same as those of other comparable transmissions, it must be realized that hydromechanical transmissions are at a very early stage of development. Fuel consumption of the test vehicles with hydromechanical transmissions was 5 to 7 percent less than that of other compatible hydrokinetic transmissions, since the hydromechanical transmission permits the engine to operate at its most economical power range. They have already proved better performance capabilities and have shown greater potential over those transmissions that had considerably longer development periods.

Designs of modular construction have been conceived which would cover all transmission ranges within the military vehicles system and would provide maximum standardization and interchanges of components.

It is therefore concluded that the development of hydromechanical transmissions will provide improvements to the capabilities of tactical wheeled vehicles. They will also improve the capabilities of tactical tracked vehicles for which two of the same units are required for power-train installation.

TREND FORECAST

Trend forecasts of present and future specific ratings of torque to weight and torque to volume of mechanical, hydrokinetic, and hydromechanical transmissions for the time frame of 1966–1980 is shown in Figs. II-1 and II-2.

Although an increase of torque-to-weight ratio for mechanical transmissions is foreseen (see Fig. II-1), the rate of increase will be less than that expected from future hydrokinetic or hydromechanical transmission developments. While the use of lighter weight materials will have the greatest impact on the weight decrease of mechanical transmissions, design innovations of the hydrokinetic and hydromechanical will account for the greater percentage of their weight savings. Hydromechanical transmissions have the greatest potential for improvement and are expected to surpass both mechanical and hydrokinetic transmissions within a few years of continued development.

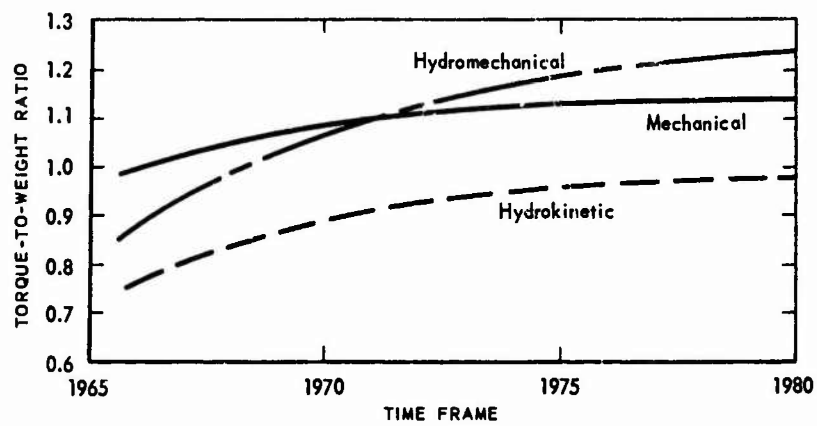

Fig. II-1—Trend-Forecast, Specific Ratios of Torque to Weight of Mechanical, Hydrokinetic, and Hydromechanical Transmissions

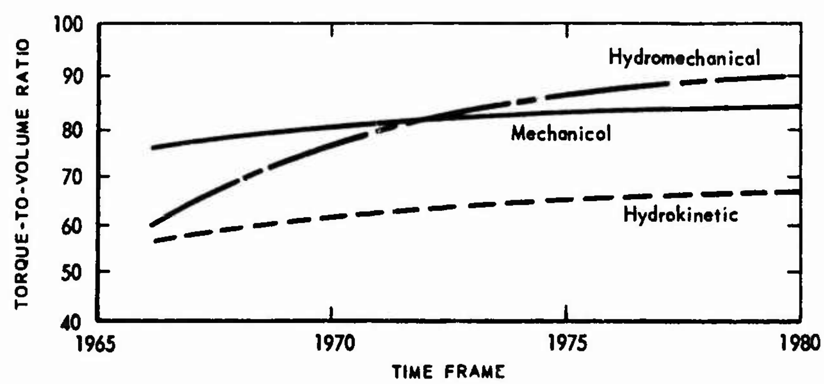

Fig. II-2—Trend-Forecast, Specific Ratios of Torque to Volume of Mechanical, Hydrokinetic, and Hydromechanical Transmissions

The expected reduction in transmission volume of future mechanical, hydrokinetic, and hydromechanical transmissions is reflected in Fig. II-2. Simplified design, fewer parts, and continued production engineering and methods will result in smaller transmissions. Further transmission volume reductions will be possible through the use of higher quality steels. As a result, some of the improvements may increase the cost, and other improvements may decrease the cost of transmissions. A reduction in the transmission size is not expected to be as significant as the reduction in weight.

Figure II-3 shows the three basic transmission types that were evaluated for wheeled-vehicle applications. It summarizes the conclusions drawn from the findings and evaluations, indicating those transmissions that have the greatest potential of improving the capabilities of tactical wheeled vehicles through Government support of R&D programs.

Type of transmission for wheeled vehicles	Findings of potential improvements through R & D			
	Yes	No	Qualified Yes	Qualified No
Mechanical:				
1. Progressive sliding gear		X		
2. Selective sliding gear		X		
3. Constant mesh		X		
4. Synchromesh				X
5. Traction drive		X		
6. Belt drive		X		
Hydrokinetic:				
1. Hydromatic		X		
2. Torque converter				X
3. Torqmatic				X
4. TX series	X			
Hydromechanical:				
1. Hydromechanical	X			

Fig. II-3—Limited Areas for Potential Transmission Improvements

SUMMARY OF MECHANICAL, HYDROKINETIC, AND HYDROMECHANICAL TRANSMISSIONS

Mechanical transmissions do not appear to have the potential of being greatly improved by continued Government development efforts. Although their operational capabilities are less than those of hydrokinetic and hydromechanical transmissions, they will continue to have application in many future tactical wheeled vehicles. Mechanical transmissions have been under development for many years and the rate of improvements from future development effort will be diminishing. In addition, improvements to mechanical transmissions will continue through development by industry, which has a prime interest in these transmissions for commercial applications. Developments for commercial applications will provide benefits for future military applications.

Hydrokinetic transmissions have been under development for a number of years and have provided the means for meeting the requirements of tactical vehicles. However, new tactical vehicle requirements have been considerably upgraded, and higher-power-range hydrokinetic transmissions are required. Since industry has very few requirements for high-power-range commercial transmissions, they have little or no incentive to develop these transmissions on their own. Therefore additional development work must be supported by the Government.

The hydromechanical transmission appears to have greater potential of improving the capability of tactical vehicles. The initially developed hydromechanical transmissions compare equally to present hydrokinetic transmissions, and their continued development should surpass the capability of the hydrokinetic transmissions. Their full development in all power ranges will probably replace many of the hydrokinetic transmissions, but this will not happen in the near future since this development effort will require at least 10 years. Military vehicle operational requirements are much greater than those for commercial vehicles, and transmission improvements are therefore more important for military than commercial applications. This fact does not provide industry with enough incentive to develop new conceptual transmissions on their own. Therefore it is concluded that hydromechanical transmissions should be fully developed by the Government for application in tactical wheeled vehicles, and due to their uniqueness they can also be applied to tactical tracked vehicles for which two of these units are required.

Chapter 19

POWER TRAINS FOR TRACKED VEHICLES

Power trains are power-conversion devices used in most tracked vehicles. They are composed of two parts, the steering portion and the transmission portion. Each complements the other, and failure of either will immobilize the vehicle.

Since much of the transmission portion of power trains has been discussed in the study of wheeled vehicles, it will not be discussed in detail here. Basic types of steering systems are presented before a general analysis of types of power trains.

STEERING SYSTEMS

Steering systems have been designed either as an integral part of the power train or as separate entities. They differ in concept and configuration according to the needs of the type of vehicle to which they are applied. A separate steering system permits vehicle designers greater flexibility in selecting the location of the steering system, whereas a system incorporated in the power-train assembly generally results in a lighter and more compact unit.

A steering system for a tracked vehicle creates a speed differential between the two tracks of the vehicle. This causes the tracks to slide transversely over the ground at different rates and changes the vehicle's direction. During this operation a power loss that affects the vehicle's speed occurs in the track on the inside of the turn. In addition to this external power loss, internal losses, other than normal mechanical-efficiency losses, occur during the steering operation. To recover some of these losses, many steering units incorporate what is commonly referred to as a "regenerative steer system." Steering units that do not have this capability are referred to as "nonregenerative steer systems." Such systems usually produce a track-speed differential by mechanically braking or declutching one track, although this may be done hydraulically. Most recently developed tracked vehicles incorporate a regenerative steer system, but nonregenerative steer systems have an application in certain types of tracked vehicles.

Many other features may be required in a steering system for tracked vehicles; systems vary greatly in concept, principle, and operating features.

The steering systems most widely used for power trains in military and commercial vehicles fall into seven types:

(a) Braked differential
(b) Clutch-brake steer
(c) Controlled differential
(d) Geared steer
(e) Clutch-brake-geared steer
(f) Hydrostatically controlled geared steer
(g) Hydrostatic steer

Vehicle capabilities provided by each type of steering system are shown in Fig. II-4.

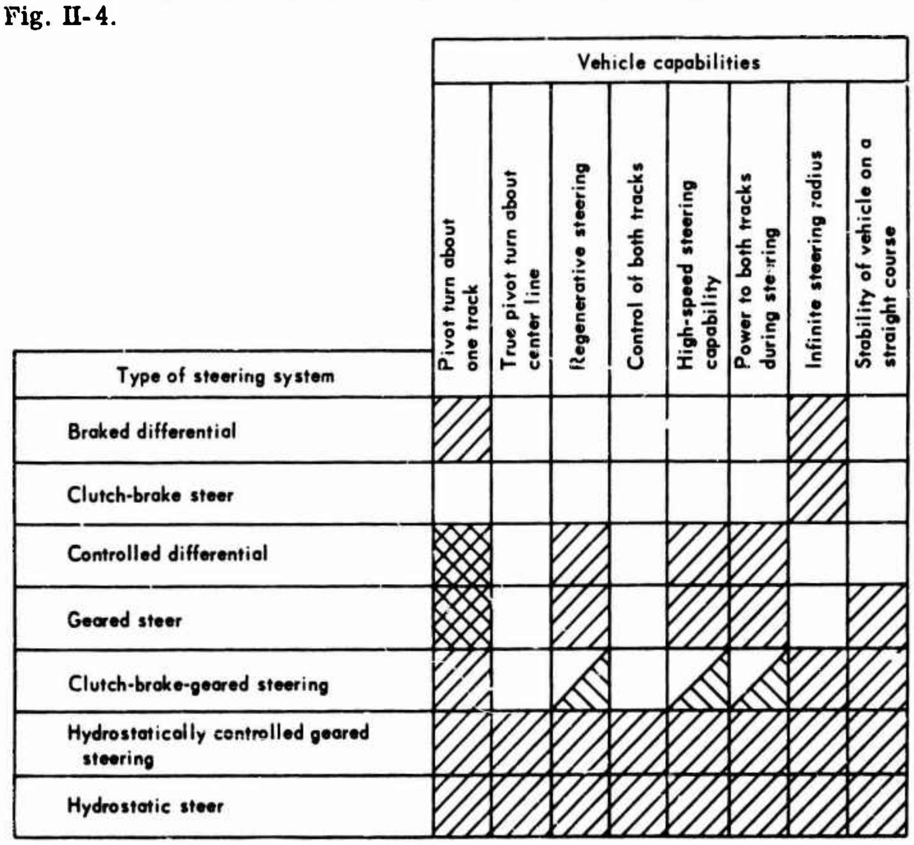

Fig. II-4—Possible Capabilities with Respect to Steering Systems

Not possible — Possible only during geared steering

Possible — Possible to some degree

Braked Differential

The braked-differential steering system is one of the earliest, simplest, and cheapest to be produced for tracked vehicles. The system is nonregenerative. Steering is accomplished by braking one output shaft, thus causing the differential to speed up the opposite output shaft. A pivot turn can be made around one track by locking its brake. Although the speed differential between the inner and outer track is potentially infinite and no gear changes are required within the power train, this type of steering system is not adequate for most military vehicles. The system makes the vehicle highly unstable on turns, and continual steering corrections are required to keep the vehicle on a straight course. As a result, brake elements wear excessively and great effort and skill are required on the part of the operator. Vehicle speed is limited by the difficulty of control. Steering the vehicle by braking one track absorbs up to 50 percent of the power normally available to the track, which prevents the vehicle from maintaining its speed. This is one reason why the system is not suitable for most tactical tracklaying vehicles. However, since it is relatively simple, cheap, and lightweight compared to more sophisticated systems, it is used for slow-moving special-purpose vehicles such as bulldozers.

Clutch-Brake Steer

The clutch-brake steer system is a modification of the braked-differential steer system, with a clutch added at each output shaft. The system is nonregenerative. Steering is accomplished by disengaging a clutch before braking the inside track. No differential is incorporated in the system, and therefore full power is transmitted to the outer track. However, power losses due to clutch and brake slippage are relatively high. Also, since one of the output shafts is disconnected during steering operations, the system cannot feed power back from the braked track.

Controlled Differential

The controlled-differential steering system incorporates two epicycle gear and brake arrangements in the right-angle drive of the rear axle, which splits power output to the drive sprockets. The operator steers the vehicle by applying the brake on one side of the system, thus causing a decrease in the speed of one output shaft and an equal increase in the speed of the opposite output shaft. The system is regenerative, and the power applied during a steering operation is proportional to the difference between inner- and outer-sprocket torques. A disadvantage is that the controlled-differential system causes the vehicle to veer off a straight course if ground conditions are not the same under both tracks. This is especially noticeable during sudden deceleration.

Geared Steer

The geared-steer system possesses the same steering characteristics as the controlled-differential system but utilizes a cross shaft to transfer speed from one output side to the other. The system is regenerative, and the power applied during a steering operation is proportional to the difference between the inner- and outer-sprocket torques.

Clutch-Brake-Geared Steer

The clutch-brake-geared-steer system combines a clutch-brake and a geared-steer principle in one system. Steering is accomplished by slipping one clutch to change the relative speed output to the tracks. At this time the system is regenerative; but when the planetary gears on one side are declutched and brakes are applied, the system is nonregenerative. A true pivot turn around the vehicle's center line cannot be made in either steering mode, but a pivot turn around one track is possible during clutch-brake operations. The steering ratio of this system is infinitely variable.

Hydrostatically Controlled Geared Steer

The hydrostatically controlled geared-steer system (or double-differential system) is similar to a geared-steer system but utilizes no clutches or brakes on the planetary gear. Full power and speed are always available at the sprockets. Steering is accomplished by using a hydrostatic motor to increase the speed of the sun gear in one planetary-gear set and proportionally decreasing the sun-gear speed in the other. This in turn causes a speedup of the planetary gears on one side and a corresponding decrease in their speed on the other. Since the planetary gears are linked to the drive shafts, this causes the vehicle to turn. A pivot turn can be made around one track, or the vehicle can be pivoted around its center line by counterrotating the tracks. The unit is fully regenerative at all times, and loss of power due to slipping clutches and brakes is eliminated. Hydrostatically controlled geared-steer systems are more sophisticated than those employing strictly mechanical principles since they provide superior steering capabilities without power losses. However, such systems are heavier, bulkier, and more complex than other steering systems.

Hydrostatic Steer

There are a number of design variations in hydrostatic-steer systems. Components and controls vary with each design, but the basic principle remains the same. The hydrostatic-steer system incorporates a pump, two motors, and a control system. The pump is driven off the power-train input shaft and in turn drives the two steering pumps located at each planetary-gear set. Different output speeds of the motors, as well as a reverse motion of one or both, can be obtained. Since the motors are geared directly to the sun gears of each planetary-gear set on the power-train output shaft, a speed differential between the sprockets can be created to steer the vehicle. This system is regenerative under all operating conditions and provides an infinitely variable steer ratio. It permits a pivot turn around one track as well as a true pivot around the vehicle's center line. The characteristics of the hydrostatic-steer system are the same as those of a hydrostatically controlled geared-steer (or double-differential) system.

POWER TRAINS

The following types of power trains for tracklaying vehicles are discussed in the following subsections:

(a) Mechanical—Constant mesh and synchromesh

(b) Hydrokinetic—"Hydramatic" (GMC trademark) and torque-converter planetary gear

(c) Hydromechanical

These power trains have been evaluated according to their physical and performance characteristics, ease of operation, flexibility for vehicle design, reliability, maintainability, and cost.

Mechanical Power Trains

Constant-mesh and synchromesh power trains incorporating either braked-differential or clutch-brake steering were used in most military tracked vehicles after WWII, and some are still in use today, but they are gradually being phased out along with the vehicles in which they are installed.

The concepts for the transmission portion of these power trains are identical to those already discussed for use in wheeled vehicles. Their construction was modified, however, to conform to the configuration and to the more demanding operating requirements of tracked vehicles. These modifications added considerable weight and size to the units.

The mechanical efficiency, simplicity, and reliability of constant-mesh and synchromesh power trains have been well established during many years of use in tactical vehicles. As speed requirements have increased, however, the operational performance characteristics of the vehicles have become increasingly inadequate. Relatively high power losses during steering and the need for continuous steering corrections have made such vehicles very difficult to control and dangerous to operate.

In most cases mechanical power trains can no longer meet vehicle performance requirements for modern tracklaying vehicles. Their use in tactical vehicles is expected to diminish as performance requirements increase. A mechanical power-train installation is usually limited to noncombat vehicles with low power and speed requirements.

Since mechanical power trains are well suited to use in commercial vehicles, industry is continually developing and improving them, along with their mechanical-transmission counterparts. This continued development will result in some decrease in unit cost. These improvements are of small importance, however, if the essential requirements of mobility and ease of operation for tactical tracked vehicles cannot be met.

A military use might be found for these power trains in new vehicles if low cost, high efficiency, ruggedness, and reliability should outweigh the requirements for mobility and ease of operation. Limited development programs to modify commercial units to meet specific tactical vehicle requirements could be justified if the configuration of the desired vehicle required greater design flexibility and small quantities were involved. If considerable vehicle production were contemplated, however, and the advantage of using a new mechanical power train became evident, a full-scale development program might be warranted.

Hydrokinetic Power Trains

The "Hydramatic" power train is a proprietary item of GMC, which has developed the system in power ranges up to 200 hp. It has been adapted for use in tactical tracked vehicles. These power trains were first installed in light and medium tanks and in several self-propelled guns a few years before WWII. Installation in a number of amphibious cargo and armored personnel carriers followed during and after the war.

The Hydramatic power train consists of a transmission, similar to the type used for wheeled vehicles, and a separate controlled-differential steering unit. A power train of this type is heavier, bulkier, and more costly than an equally rated unit of integral construction but may be desirable for vehicle-design flexibility. Since these power trains have limited power capability, two units were needed in each light and medium tank. This arrangement is heavier, bulkier, more expensive, and considerably less efficient than a single unit with sufficient power capability. However, it does have the ability to "get home" if one unit should fail.

Improved controllability at high speeds and the greater ease of operation of the Hydramatic power-train system increased the mobility of tactical tracked vehicles using this system over those using mechanical power trains. These power trains were not developed for all required power ranges, however, since other more promising power trains were in the process of development. This type of power train in military use has been superseded by various versions of improved torque-converter planetary-gear power trains. The latest and most widely accepted version is the X series.

All modern power trains for tracked vehicles in the military system employ hydraulic torque converters and hydraulically controlled planetary gearing combined with hydraulically operated steering systems. Development of these power trains was initiated at the beginning of WWII, and they were first installed in vehicles in 1943. Although the first units were inefficient and bulky by today's standards, their performance was superior to that of previously installed power trains. Subsequent development programs have overcome the deficiencies of the early models, and a greater selection of power ranges is now available for tracked vehicles.

Figure II-5 shows the various types of hydrokinetic power trains used in military tracked vehicles, with their respective steering systems, in the order of their development. The chart indicates the vehicle-weight classes and torque ranges in which these power trains are at present (1966) available. It can be seen that none of the available power trains covers the entire torque range for all military tracked vehicles. Consequently for some vehicles either a lower-rated unit, which would reduce operating life, or a higher-rated unit, which would increase the vehicle's weight, must be selected. In addition, some power trains do not have the proper configuration or operating characteristics for a particular vehicle, even if the proper torque range is available. This is the primary reason why power trains are available in many different models.

The objectives of development programs for hydrokinetic power trains have been to improve the mobility of the vehicle and its ease of operation; to provide greater reliability and efficiency, to increase ease of maintenance; and to reduce weight, size, and production cost. Most of these objectives have been met in the design of the X series power trains. Figures II-6 and II-7 show the progressive specific-rating improvements attained during the development of hydrokinetic power trains with respect to torque-to-volume and torque-to-weight ratio. Although power trains of the X series have increased the capabilities of tracked vehicles, their potential has not been fully exploited. Further development is needed to improve the reliability, ease of maintenance, efficiency, weight, size, and unit cost of this power train. as well as to enable it to meet

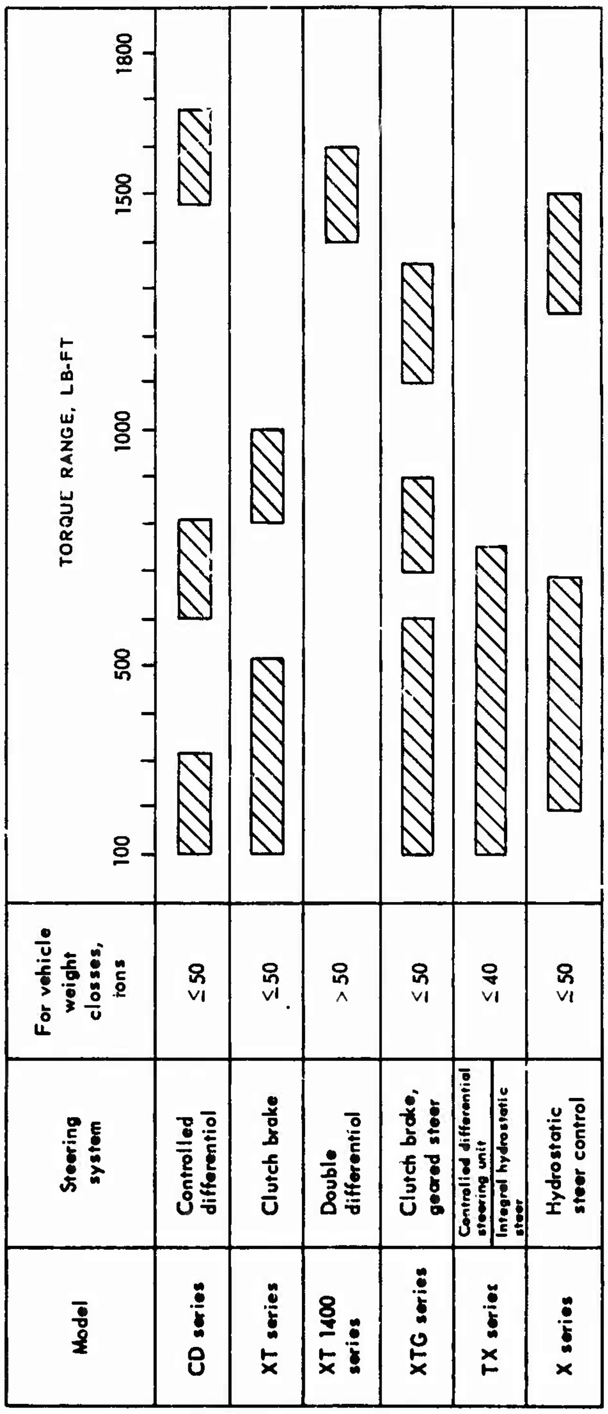

Fig. II-5—Power-Train Usage and Torque Range all torque-range and configuration requirements. It is estimated that 10 years of development with full Government support would be required to achieve these improvements.

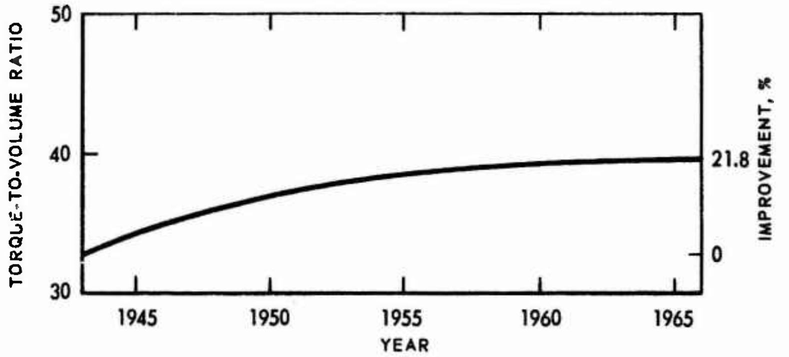

Fig. II-6—Specific Torque-to-Volume Improvements during Hydrokinetic-Power-Train Development Period

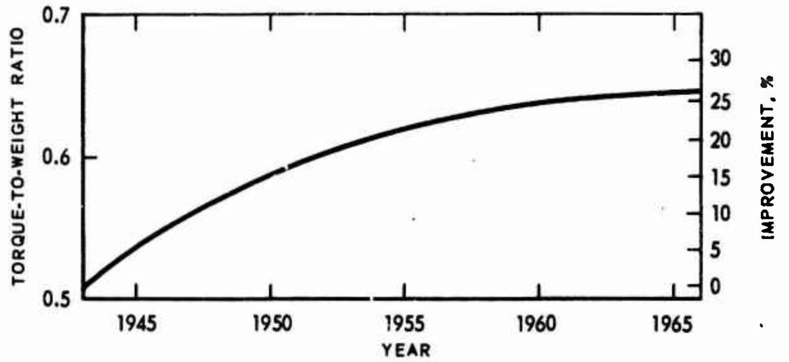

Fig. II-7—Specific Torque-to-Weight Improvements during Hydrokinetic-Power-Train Development Period

Hydromechanical Power Trains

The hydromechanical power train has been under development since 1962 by the General Electric Company, Schenectady, N. Y., and by the Hydromechanic Research Center of the American Brake Shoe Corporation, Columbus, Ohio, under several Government contracts. The power train was installed in a tracked vehicle (XM104) and, after some modifications, yielded very favorable test results. This system appears to have the greatest potential for improving the performance of future tactical tracked vehicles.

The construction of the hydromechanical power train is similar to that of a hydromechanical transmission except that two units are used for the power train. Since individual hydrostatic controls are used in each unit to vary the torque and speed at each output shaft, the power train has inherent steering capabilities.

Although hydromechanical power trains do not appreciably improve vehicle mobility over that of vehicles incorporating the most recently developed torque-converter planetary-gear power train, they offer improved specific ratings, acceleration, and fuel economy. Fuel consumption in a test vehicle with a hydromechanical power train was between 5 and 10 percent less than with a comparable hydrokinetic power train, because the hydromechanical system permitted the engine to operate in its most economical power range. The torque-to-weight ratio was determined to be approximately 45 percent higher, and the torque-to-volume ratio was determined to be approximately 28 percent higher than that of a comparable hydrokinetic power train. Ease of maintenance and reliability appeared to be greater since components were easily removable and 25 percent fewer parts were required than for hydrokinetic transmissions.

It was determined, however, that because of less efficient fluid mechanics the overall efficiency of the hydromechanical power train was less than that of hydrokinetic or purely mechanical power trains. A comparison of their efficiency is shown in Fig. II-8.

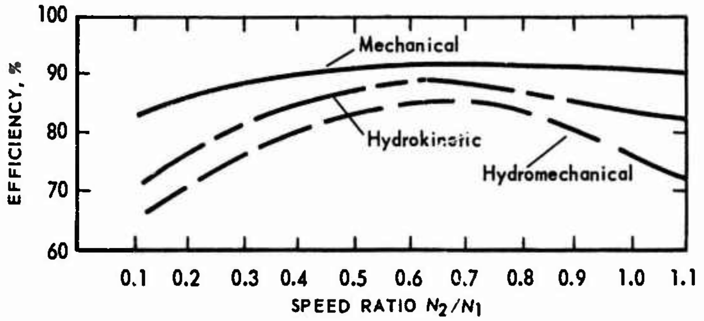

Fig. II-8—Comparison of Efficiency of Mechanical, Hydrokinetic, and Hydromechanical Power Trains

N_1 = engine output RPM; N_2 = power train output RPM.

The hydromechanical power train offers several advantages not present in other power trains. No brakes or clutches are required for steering, and engine output requirements and fuel consumption are reduced by efficient regenerative steering. Operator efficiency is improved, and the system is adaptable to various engine types since no torque converter is used.

Hydromechanical designs of modular construction have been conceived that could cover all power ranges for transmissions and power trains used in tactical vehicles, and that would provide maximum standardization and interchangeability of parts. A program to develop this concept would be extremely costly and would require at least 10 years of effort. Present plans are to incorporate components that would permit the hydrostatic system to operate at higher pressures, thus increasing the capacity of the power-train unit without appreciably increasing its weight and size.

The development trend forecast shown in Figs. II-9 and II-10 predicts the specific ratings of the three basic types of power trains during the time frame 1966–1980. Acceleration will improve with faster-reacting controls, which will also provide better fuel economy by avoiding unnecessary time lags at unfavorable engine speeds. Improvements in reliability and maintainability will be suggested to the designer through the actual use of hydromechanical power trains in operating vehicles. These improvements will increase the maneuverability and agility, and therefore the effectiveness, of tactical tracked vehicles.

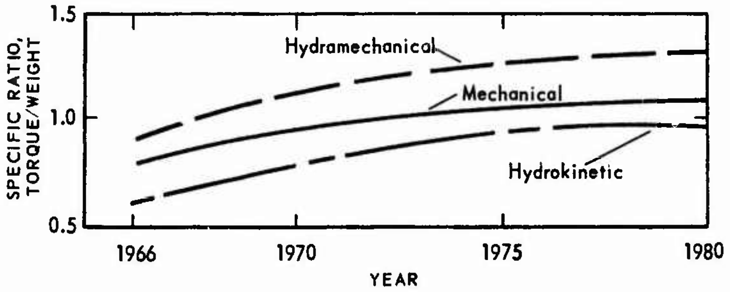

Fig. II-9—Power-Train Development Trend (Torque/Weight)

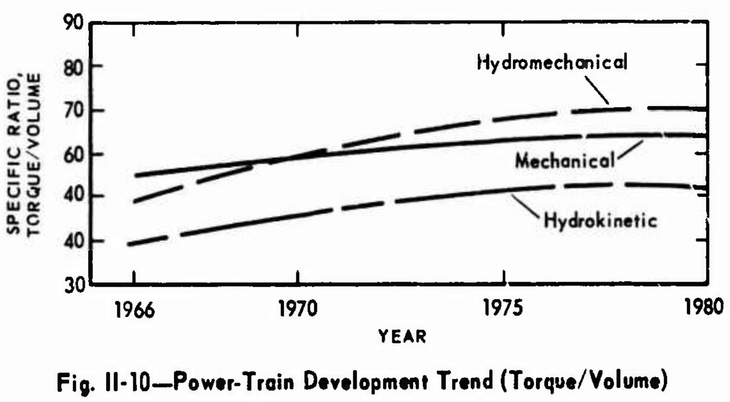

Fig. II-10—Power-Train Development Trend (Torque/Volume)

R&D will be carried out by industry on a few commercial applications of hydromechanical systems, and future military development will benefit as a result. But full military developmental effort is required to exploit the full potential of hydromechanical power trains and to extend their power range to cover a wide variety of tactical tracked vehicles.

Figure II-11 summarizes the potential of the three basic types of power trains to improve the performance of tactical tracked vehicles through Government support of R&D programs.

Type of power train for tracked vehicles	Findings of potential improvements through R & D			
	Yes	No	Qualified	
			Yes	No
Mechanical				
1. Constant mesh				X
2. Synchromesh				X
Hydrokinetic				
1. Hydramatic		X		
2. Torque-converter planetary gear, CD series		X		
3. XT series		X		
4. XTG series		X		
5. TX series		X		
6. Torque-converter planetary gear, X series	X			
Hydromechanical				
1. Hydromechanical	X			

Fig. II-11—Limited Areas for Potential Power-Train Improvements

SUMMARY

Mechanical Power Trains. Present mechanical power trains are not fulfilling the requirements for tactical tracked vehicles. Industry is developing mechanical power trains for commercial applications, and these units could be modified, if necessary, for military use in specific tactical vehicles. Therefore Government development of new mechanical power trains cannot be justified.

Hydrokinetic Power Trains. Hydrokinetic power trains have generally replaced mechanical power trains for use in tactical tracked vehicles. The latest development is the torque-converter planetary-gear power train with integral hydrostatic steering, designated as the X series. This unit is reasonably efficient and compact and fulfills most tactical tracked-vehicle requirements. It does not cover all power ranges for all classes of tracked vehicles, however, and present units have additional improvement potential. Government developmental effort is required on the X series power train to provide the full span of power ranges that will be required within the next 10 years, or through 1975.

Hydromechanical Power Trains. Hydromechanical power trains have great potential for improving the performance of tactical tracked vehicles, but they are not yet available for production vehicles. Only a few prototype units have been produced, and additional testing will be required before drawings and tooling for initial production can be produced; even then, only one power range will be available. It is estimated that at least 10 years of intensive developmental effort will be required to ensure that reliable production units will be available in all desired power ranges. Full Government support is required to develop hydromechanical power trains for use in tactical tracked vehicles fielded in the 1975–1985 time frame.

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Chapter 20

HYDROSTATIC DRIVES

BACKGROUND

A hydrostatic drive is a power-conversion device that transmits power through the flow of hydraulic fluid from the power source to the wheel or sprocket of a vehicle. A hydraulic pump forces the fluid to a motor at regulated pressures.

A hydrostatic-drive system in a wheeled vehicle employs a motor for each driving wheel and is called a "hydrostatic-drive transmission." A hydrostatic-drive system in a tracked vehicle employs a motor for each driving sprocket, with a differential speed-control system providing the means of steering the vehicle. This type of system is called a "hydrostatic-drive power train."

Hydrostatic drives offer continuously variable output speed, dynamic braking, design flexibility in the location of hydrostatic-drive components, and ease of control in operating the vehicle. For these reasons they are of great interest to vehicle designers.

During the early stages in the development of the hydrostatic-drive system, attempts were made to utilize readily available shelf components. This approach resulted in a system that was heavy, bulky, inefficient, and unreliable. To overcome these deficiencies, industry conducted R&D programs to develop components that would increase the capability of hydrostatic-drive systems to operate at higher pressures.

In early positive-displacement gear pumps or motors (Fig. II-12), the volume of hydraulic oil was varied only by input speed. Hydraulic operating pressures could not exceed 2000 psi due to unbalanced fluid forces, and speeds were limited to 2000 rpm.

Later developments produced vane pumps or motors (Fig. II-13), which were more reliable and whose displacement could be varied. This made it possible to vary the volume of fluid flow while the input speed remained constant, but due to unbalanced fluid forces these units were rated for maximum pressures at 2500 psi and maximum speeds of 2000 rpm.

Subsequent developments produced ball-piston pumps or motors (Fig. II-14), which permitted operating pressures to be increased to 3250 psi and speeds to 4000 rpm.[1,2] Displacement could be varied and a controlled variable output speed could be obtained from a constant input speed, but the system had poor efficiency at high speeds and pressures.

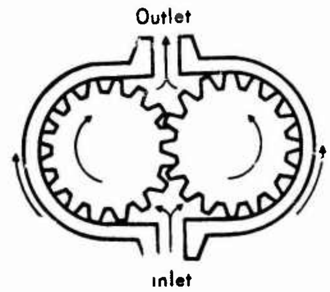

Fig. II-12—Typical Positive-Displacement Gear Pump or Motor

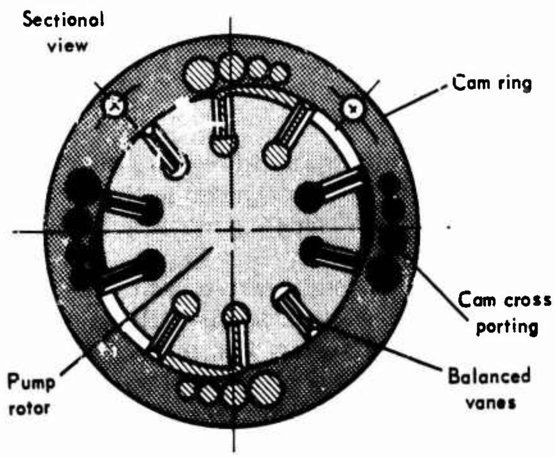

Fig. II-13—Typical Variable-Displacement Vane Pump or Motor

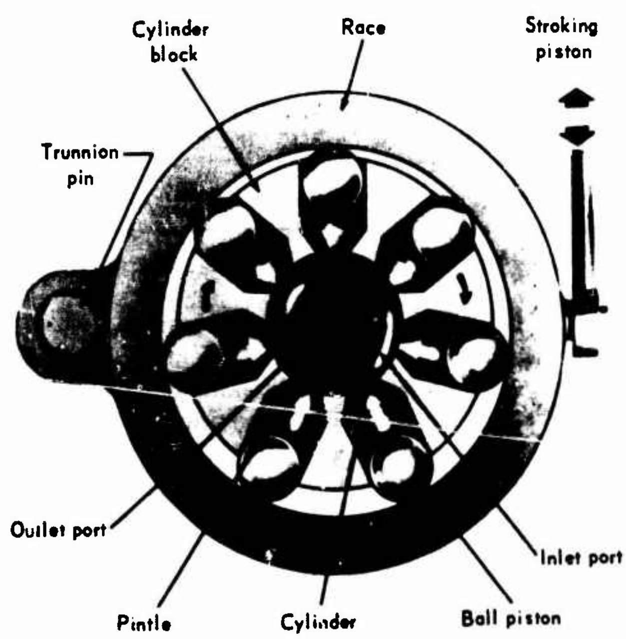

Fig. II-14—Variable-Displacement Ball-Piston Unit

One of the latest developments produced axial-piston pumps or motors that increased operating speeds to 4000 rpm and operating pressures to 6000 psi. To obtain varied displacement and output torque ratios, two different methods were developed: The first incorporated a nonrotating driving disk, called a "swash plate," whose angle of tilt in the pump or motor could be varied (see Fig. II-15). The second incorporated a rotating drive disk constructed so that the entire head could be tilted around the transverse axis to produce piston motion (see Fig. II-16). This arrangement eliminated the use of sliding slippers.

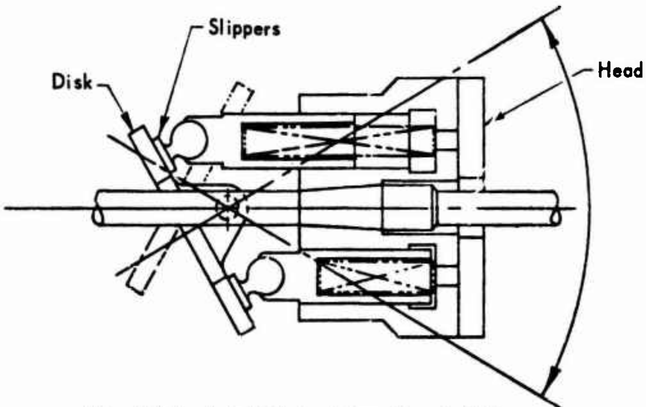

Fig. II-15—Axial-Piston-Type Swash Plate

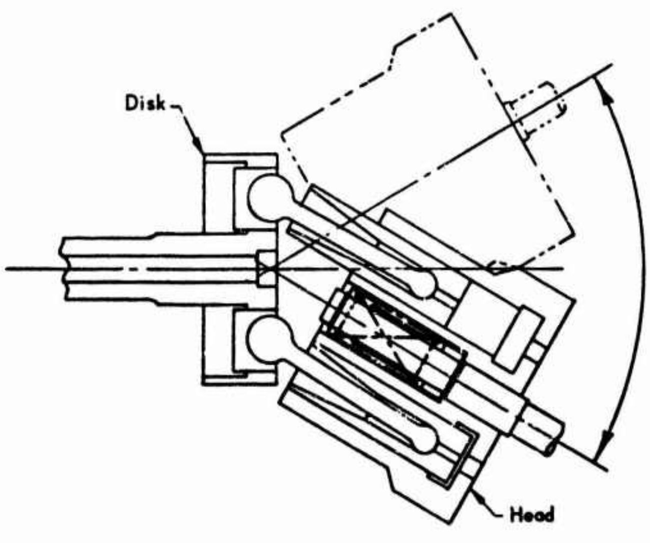

Fig. II-16—Axial-Piston-Type Tilting Head

The second method is more reliable and more efficient and can operate at wider speed and torque ranges, but it is larger and heavier than a unit incorporating the first method, since the housing must accommodate the tilting head.

A new concept is being developed to produce a vane pump with pivoting tips on the vanes (see Fig. II-17).[3] The tips bear hydrodynamically against the

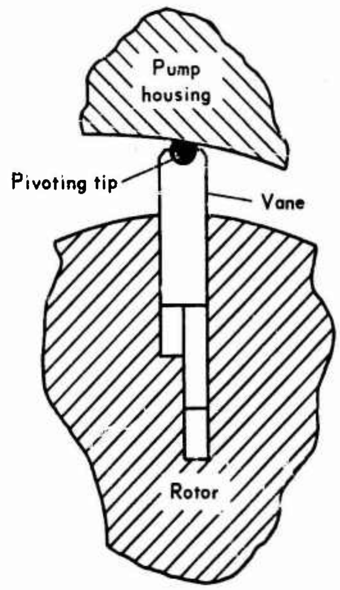

Fig. II-17—Section of a Pivoting-Tip Vane Pump[3]

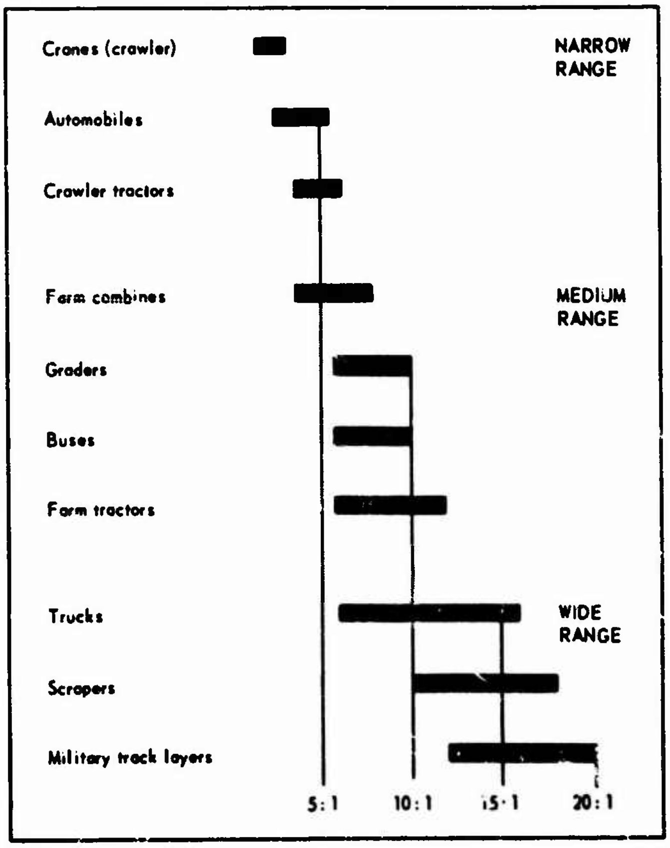

Fig. II-18—Various Vehicle-Speed Torque-Ratio Requirements[1] pump housing to seal against leakage. The pump's dual rotors produce balanced fluid forces. Prototype pumps have operated at speeds up to 30,000 rpm and pressures up to 8000 psi. These high operating speeds could permit the direct coupling of a hydrostatic pump to a gas turbine engine.

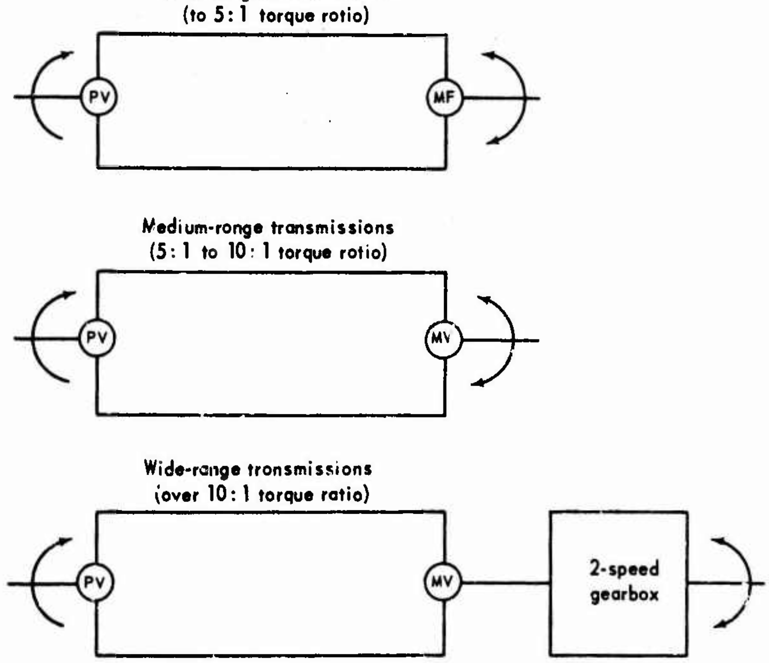

Fig. II-19—Typical Hydrostatic Pump-Motor Arrangements[4]
PV, pump, variable displacement.
MV, motor, variable displacement.
MF, motor, fixed displacement.

Speed-to-torque ratio requirements vary according to type of vehicle (see Fig. II-18).[4] Typical hydrostatic pump-motor arrangements (see Fig. II-19)[4] illustrate the type of system that would be required to meet specific speed-to-torque ratio requirements. The wide-range system requires the use of a 2-speed gearbox. This gearbox would add weight and require more space, additional controls, and a method of matching torque to achieve full-power shifting. Therefore a hydrostatic drive is more suitable for vehicles that operate in narrow to medium speed-to-torque ratio ranges.

COMMERCIAL APPLICATIONS

In recent years industry has devoted considerable effort to R&D of hydrostatic-drive transmissions. This has resulted in their use in commercial vehicles such as garden tractors and lawn mowers and as front-wheel-drive assists

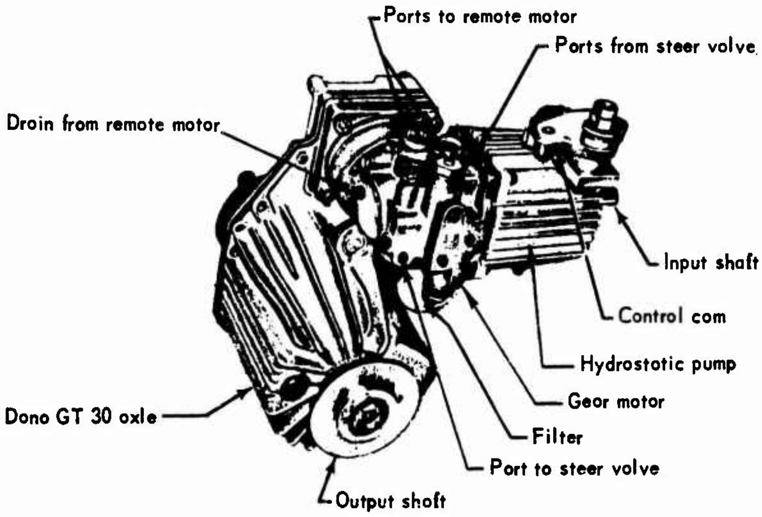

Fig. II-20—Typical Integral Hydrostatic-Drive Tran. mission
Volume, 1.5 ft^3; weight, 115 lb.

TABLE II-1

Component Manufacturers for Hydrostatic-Drive System[7]

Company	Displacement per revolution, in.^{3}	Rated pressure, psi	Rated speed, rpm	Output torque, lb-in	Input power, hp
The Dynex Co.	0.6	6000	3000	540	25
	9.0	6000	2400	8,000	300
Eaton (Dowty) Automotive Gear	0.256	1400	3600	636	10
Div. of Eaton,Yale & Towne, Inc.	12.41	4000	2500	7,560	80
Gar Wood	7.24	3000	5000	4,480	250
	51.8	2500	2500	21,875	1000
Hydreco Div. of New York Air	4.83	5000	3000	3,600	60
Brake Co.	12.0	5000	3000	8,220	225
Oilgear Co.	2.40	2500	1800	900	25
	68.0	2500	900	33,000	450
Rockwell Mfg. Co.	15.0	500	1800	1,190	40
	165.0	500	1200	13,150	300
Sundstrand Corp.	30.0	5000	1400	24,000	250–300
	2.0	5000	3800	1,600	30–40
Ulrich Mfg. Co.	2.0	5000	6000	2,000	30
	15.0	5000	4600	25,000	250
Vickers Inc. Div. of Sperry	23.0	5000	1800	18,300	525
Rand Corp.	0.65	4000	4000	715	14
Fairchild Stratos Div. of Fairchild Hiller Corp.	2.665	4500	6000	14,100	50

in farm tractors and in similar low-speed vehicles.[5,6] Most of these vehicles operate within narrow torque and speed ranges. Since industry's prime objective is to manufacture products at competitive costs, efficiencies as low as 70 percent are acceptable. Hydrostatic drives have permitted industry to design vehicles with smooth stepless control within their speed range. They have also permitted greater flexibility in vehicle design and have provided a means of improving the operator's ease of control.

A typical integral hydrostatic-drive transmission installed in a garden tractor is shown in Fig. II-20. This drive incorporates a variable-displacement axial-piston pump and gear motor. The hydraulic drive and trans-axle weigh 115 lb and occupy 1.5 ft^3, for a rating of 12 hp.[6] The performance of the hydrostatic-drive garden tractor is superior to that of a mechanical-drive garden tractor.

At present, several manufacturers are producing hydrostatic components for vehicle installation, since many commercial applications exist. Many of these components are suitable for special types of tactical vehicles. Table II-1[7] lists some of the more prominent manufacturers with their respective ranges of available components.

MILITARY APPLICATIONS

Several prototype hydrostatic drives have been installed in tactical vehicles. Early installations proved heavier, bulkier, and less efficient than the mechanical transmissions they replaced, and lacked a good steering system and adequate control circuits. These deficiencies were basically due to the use of shelf components that were available at the time. These early vehicle installations included the LARC XV, the M41, and the 6 × 6 truck. They provided valuable information to the design engineer and made possible the design of better components and more effective control circuitry.

Recent installations of hydrostatic drives in vehicles have greatly improved efficiency, weight, size, and steering controls and have permitted high maneuverability at slow speeds while retaining stability at high speeds. These installations are shown in Table II-2.

Typical hydrostatic-drive systems that have been installed in tactical vehicles are shown schematically in Fig. II-21. These installations are either integral, split, or separate hydrostatic-drive systems.

The integral-drive system incorporates pumps, motors, differential, controls, brakes, and other accessories in a common case with common manifolding. The only oil-line connections required are to and from the oil cooler (see Fig. II-22).

The integral-drive system was used in a prototype marginal terrain vehicle named the "Turtle," designed and built by USAERDL. The hydrostatic drive[8] did not have sufficient capacity for this application, however, and prototype testing was discontinued. Another integral-drive installation was made in the Universal Engineering Tractor (UET).[9] The UET (Fig. II-23) is a tracked vehicle that performs as a bulldozer, grader, earth mover, and a cargo and personnel carrier. The vehicle's many functions require high tractive effort at low

TABLE II-2

Typical Recent Hydrostatic Drive Installation in Tactical Vehicles

Vehicle installations	Drive type	Present status	Remarks
BEST (wheeled)	Split	Under test	This installation reduced the weight and bulk of the existing system; performance was satisfactory, but efficiency was low, and some maintenance problems developed
Marine Tow tractor (wheeled)	Split	In production	A good application for hydrostatic drive system since the overall height was reduced and controllability was improved
ATAC all magnesium 5-ton vehicle (wheeled)	Split	Unknown	Data are not available on this application
USAERDL Turtle (tracked)	Integral	Testing discontinued	This vehicle was one of a kind to demonstrate marginal-terrain capabilities; the integral transmission was overstressed and failed; no further testing is scheduled
TR14 transporter (wheeled)	Split	Testing continuing	This is a good application for a hydrostatic-drive transmission due to configuration of vehicle; the performance of this slow-moving vehicle is good
Howitzer: M123A1, X123E1, M124E2 (wheeled)	Split	Under test and evaluation	These are kits to provide auxiliary propulsion for the howitzers; this is a good application for hydrostatic-drive transmission since weight and volume are held to a minimum
UET (tracked)	Integral	Under test and evaluation	This installation reduced overall weight, provided excellent control, and increased torque range at dozing speeds
XA-20B (tracked)	Separate	Testing completed	This installation was made to make a direct comparison with the hydromechanical transmission; the performance was satisfactory but overall weight was increased

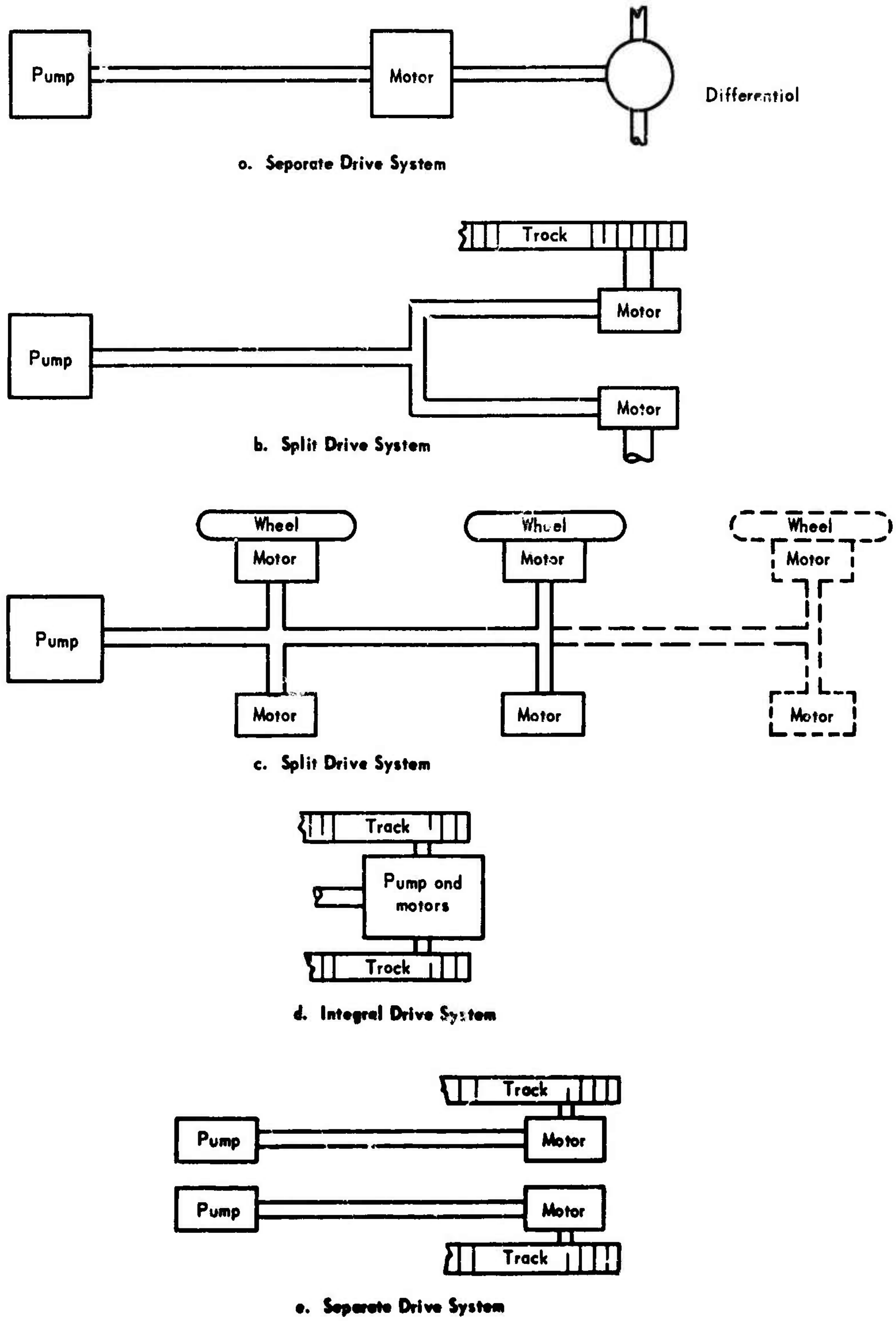

Fig. II-21—Typical Hydrostatic-Drive Systems

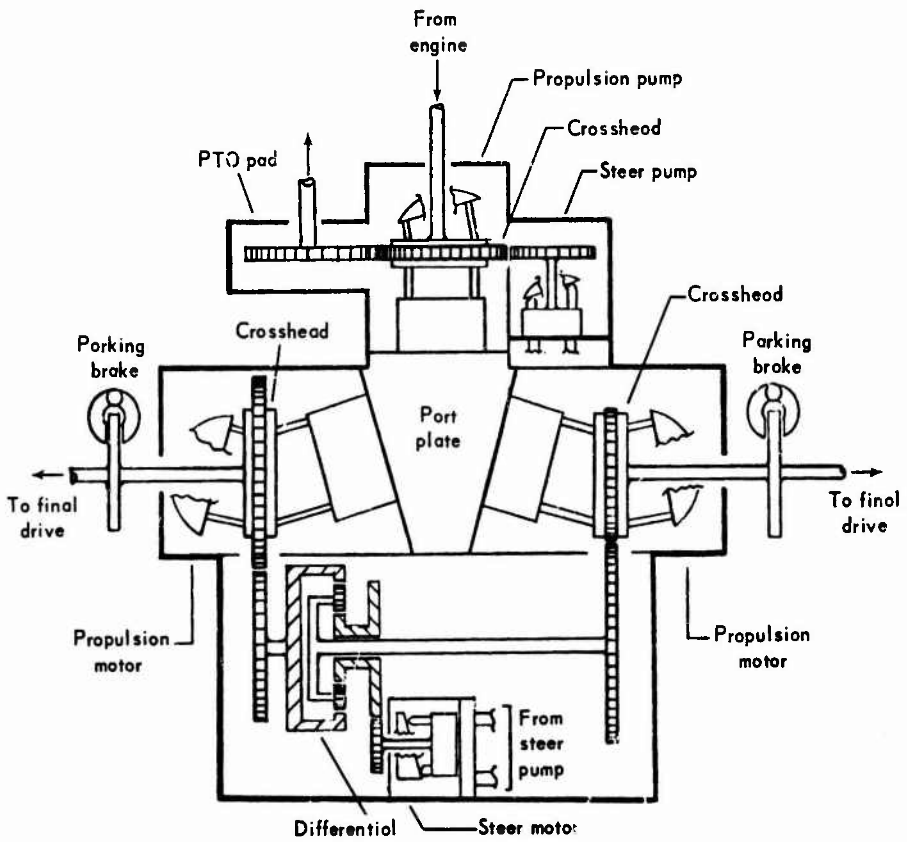

Fig. II-22—Integral-Drive Power Train

Fig. II-23—Universal Engineer Tractor

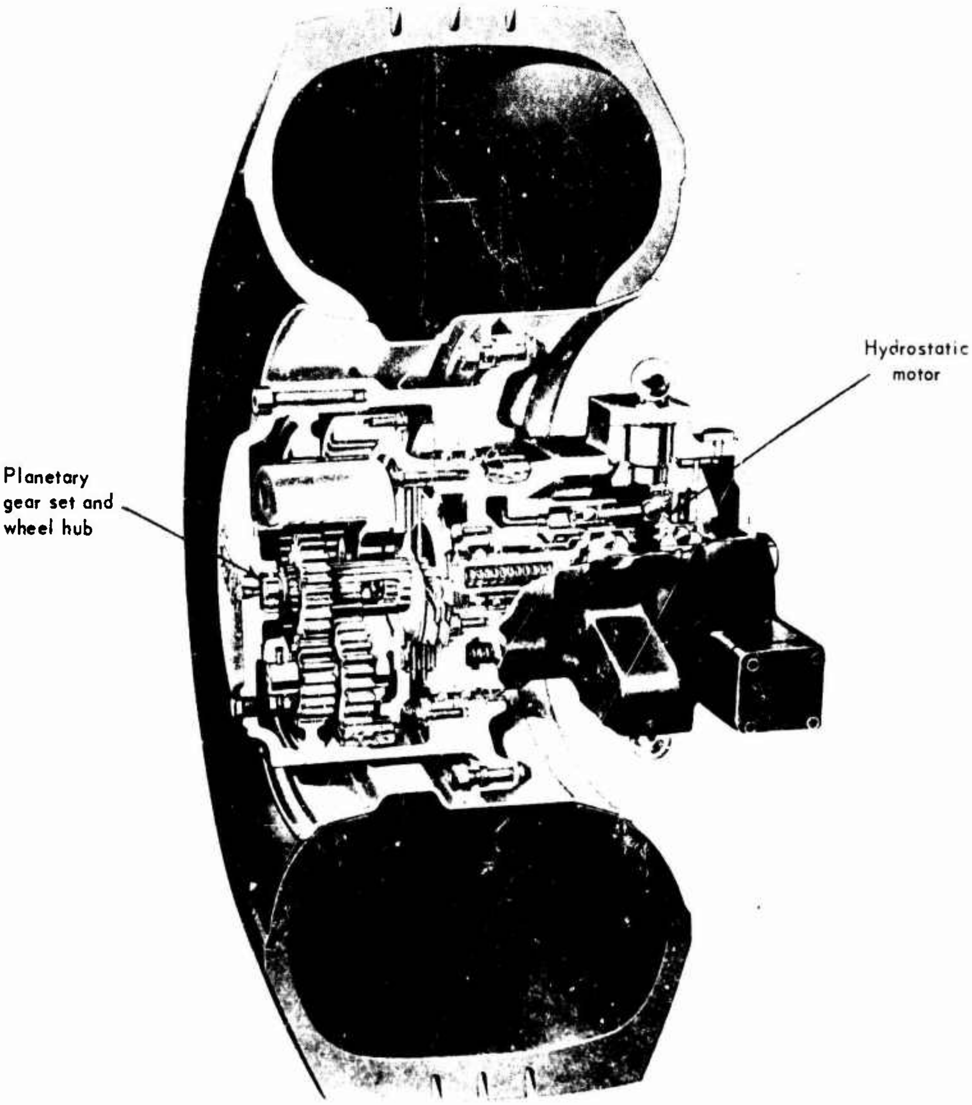

Fig. II-24—Typical Hydrostatic Wheel Drive Unit

speeds for dozing and grading, and low torque and high speed for use as a personnel or cargo carrier. The integral hydrostatic-drive power train weighs approximately 500 lb less than the hydrokinetic power train it replaced. Although service testing is only in its initial stages, preliminary results are encouraging, and indications are that most of the vehicle's operating requirements will be met.[10]

Split and separate drive systems incorporate pumps and motors in individual housings, interconnected by separate piping that carries the hydraulic fluid. The individual-wheel drive system, in which the hydrostatic motor is located in the wheel hub, is an example of a split drive system (see Fig. II-24).[11]

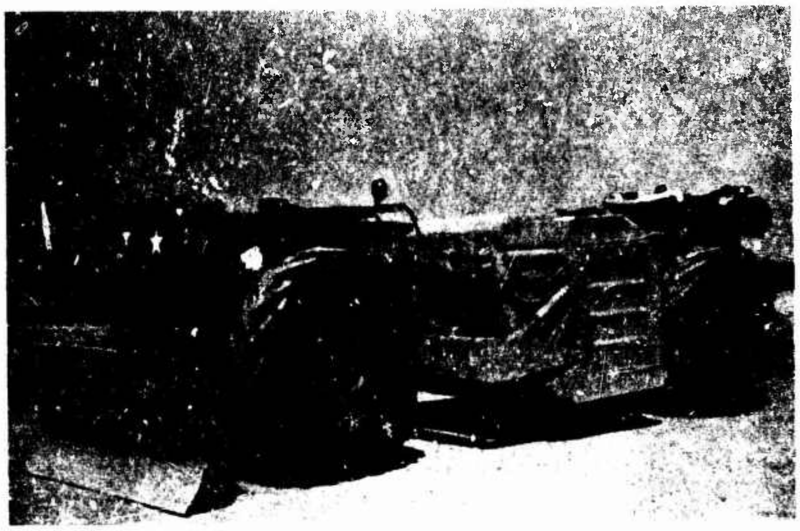

Fig. II-25—Ballastable Sectionalized Tractor

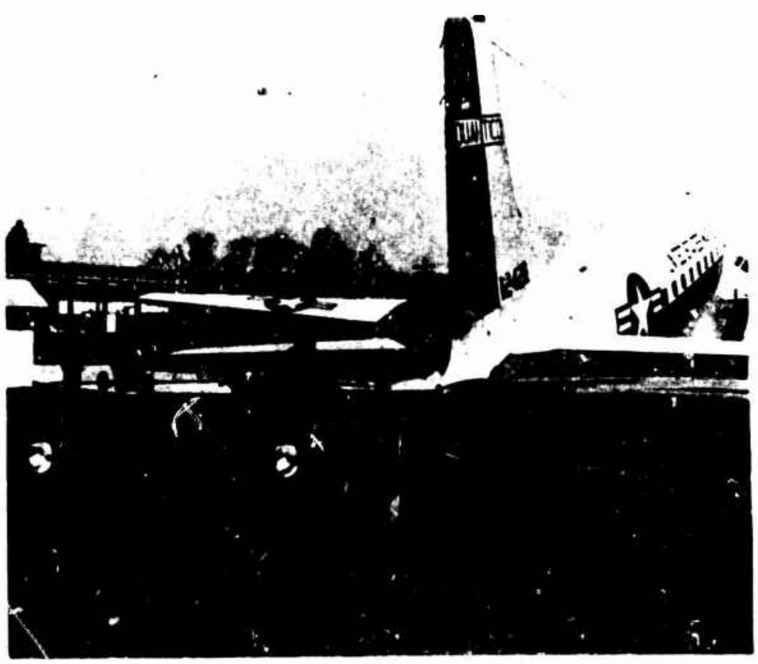

Fig. II-26—Marine Tow Tractor

An installation of this type has been made in the Ballastable Sectionalized Tractor (see Fig. II-25). This vehicle is an articulated wheeled vehicle that needs a front-wheel powered assist when high tractive effort is required. The front wheels originally were powered electrically, but hydrostatic units were used to reduce weight and volume requirements. Although the vehicle has performed satisfactorily, its efficiency is marginal, and some maintenance problems have developed.[10]

Another split-drive installation was made in the Marine Tow Tractor TD 4595,[12,13] which is now (1966) in production. The Tow Tractor is a defensive, extremely low-profile vehicle (see Fig. II-26) that is required to exert high tractive effort at relatively low speeds. The hydrostatic drive provides traction to all four wheels during low-speed operation and to only two wheels during higher-speed operation. The overall height was reduced by 6 in. by replacing the vehicle's mechanical drive system with a hydrostatic unit.

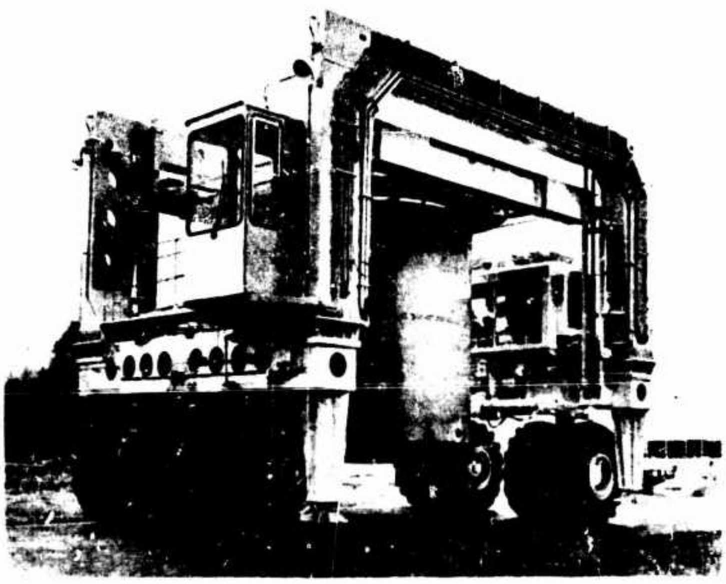

Fig. II-27—TR-14 Transporter

A split-drive installation that proved successful was made in a TR-14 Transporter (Fig. II-27).[14] The vehicle was required to operate at slow controlled speeds in four directions. This made it necessary for individual wheels to be powered, which could not have been accomplished by a mechanical power train.

Split hydrostatic drives have been installed in several models of 105- and 155-mm howitzers to make them self-propelled when separated from their prime movers (see Figs. II-28 and II-29).[15,16] This could not have been accomplished by mechanical means since the driving motors are remotely located at the wheels, and the engine and pump had to be located at the trails for proper weight distribution.

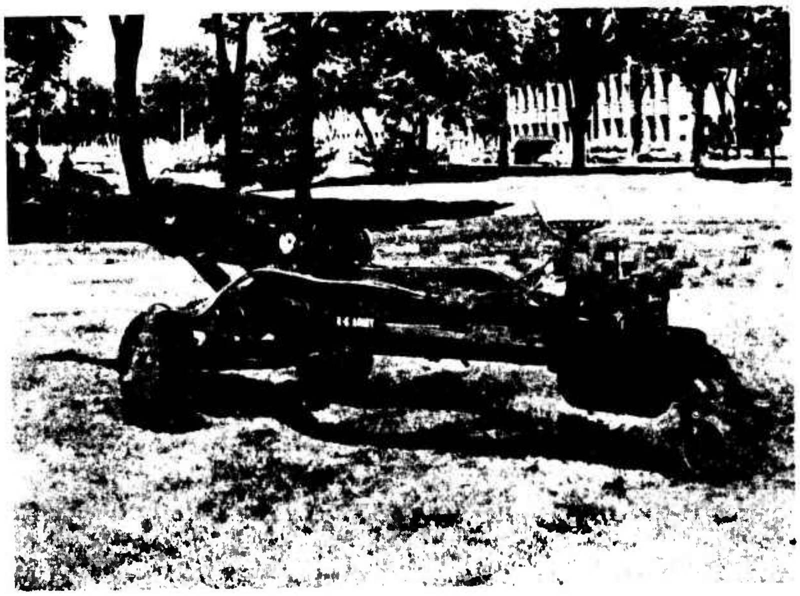

Fig. II-28—Self-Propelled 105-mm Howitzer

Fig. II-29—Self-Propelled 155-mm Howitzer

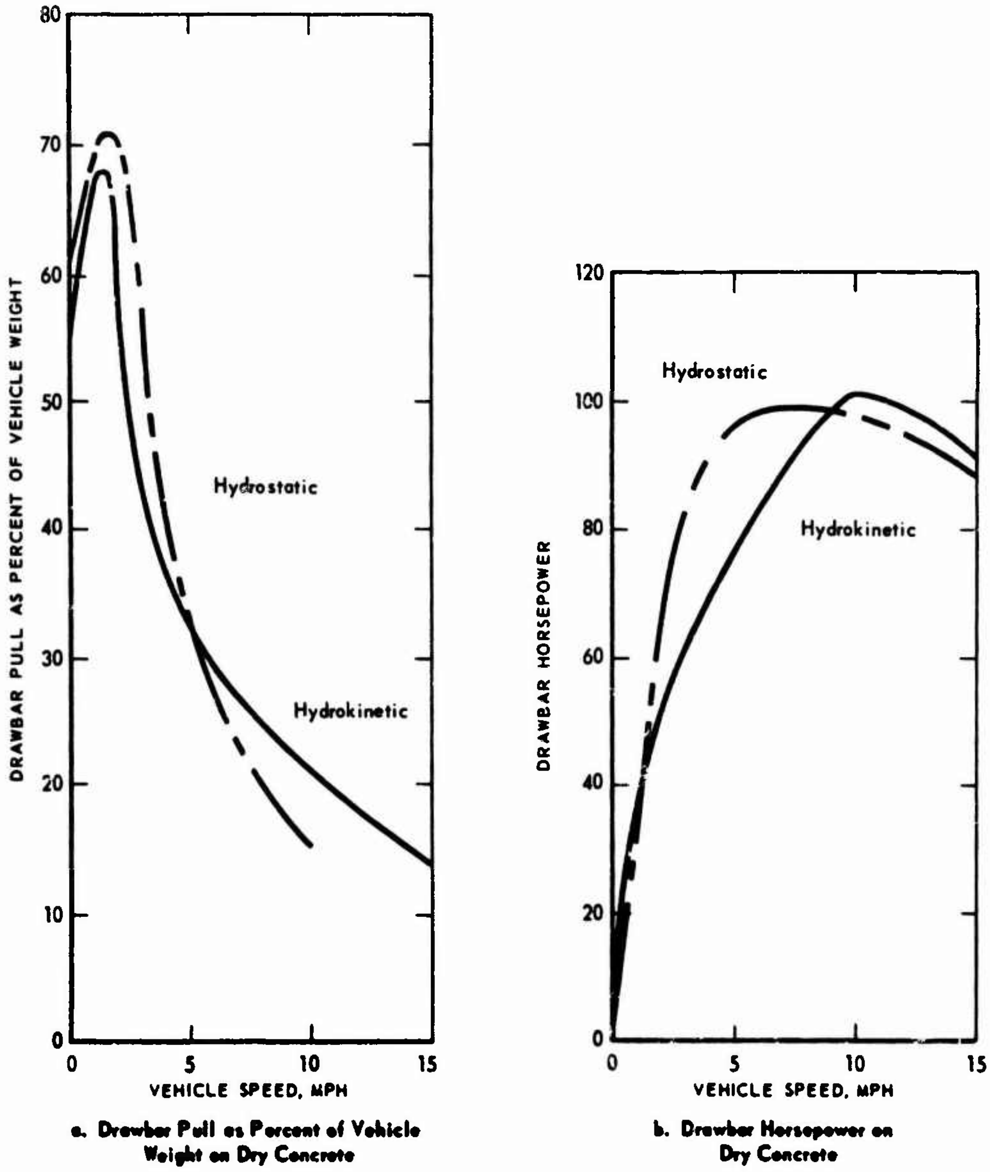

Fig. II-30—Comparison of Test Results for Hydrostatic and Hydrokinetic Power Trains

A separate hydrostatic-drive power train was installed in a Canadian tactical tracked vehicle (XA20B) to obtain a comparison between similar hydrokinetic- and hydrostatic-drive power trains. Hydrostatic drive increased the vehicle's weight by 3240 lb, but overall performance was improved. Test results for drawbar pull and drawbar horsepower are shown in Fig. II-30.[17] Although hydrostatic drive compared very favorably with the hydrokinetic power train, a gearbox had to be installed to obtain the speed-torque range required for this application.

The use of hydrostatic-drive systems in tactical vehicles has made it feasible to design vehicles with physical and performance characteristics not previously attainable.

The designer may locate various components at optimum vehicle locations. The system can readily provide independent power to all drive wheels or sprockets, and the operator can control the vehicle with greater ease. Mobility is improved, since all wheels or tracks can provide traction at infinitely variable torque and speed ranges. In some special vehicle applications, hydrostatic drives have resulted in lighter and smaller vehicles as in the case of the UET. Installation of hydrostatic drive in this vehicle reduced the weight and size of the power train by 22 and 17 percent, respectively, over the hydrokinetic power train that it replaced. However, this degree of weight and size reduction may not be obtainable for all types of vehicles. Also the hydrostatic-drive power train proved less efficient than the hydrokinetic power train in most of the output-to-input speed ratios (see Fig. II-31).

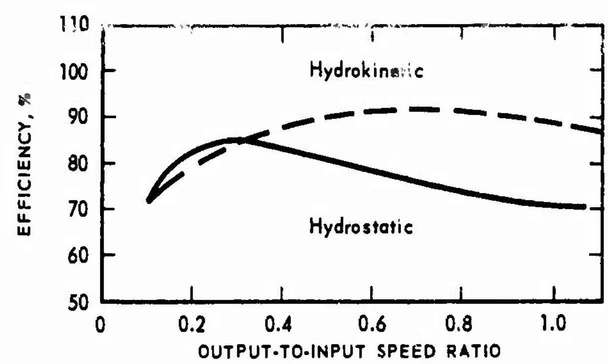

Fig. II-31—Comparison of Hydrostatic-Drive System with the Hydrokinetic-Power-Train System

Since the number of vehicle installations is limited, reliability and maintainability for the hydrostatic-drive system could not be precisely determined. However, the hydrostatic system in the Marine Tow Tractor demonstrated better reliability and maintainability than the mechanical power train it replaced.

There are numerous advantages and disadvantages in the use of hydrostatic drives for tactical vehicles. Some of the advantages of hydrostatic drives are:

(a) Designer is allowed greater flexibility in meeting the vehicle's physical and performance characteristics.

(b) Operator is enabled to vary vehicle speed independently of engine speed.

(c) Smooth continuously variable speed control is provided.

(d) Operator has ease of control.

(e) Dynamic braking is provided.

(f) Full speed and torque ranges in reverse are provided.

(g) Method of transmitting power to all wheels is provided when it cannot be accomplished satisfactorily by mechanical means.

Some of the disadvantages of the hydrostatic drive in tactical vehicles are:

(a) System efficiencies are lower than those in other comparable mechanical or hydrokinetic power trains.

(b) Speed and torque ranges are limited.

(c) Noise levels are high at elevated operating pressures.

(d) Cost of system is two to three times higher than that of present hydrokinetic power trains.

An additional disadvantage is that higher operating pressures, desired to increase specific torque-to-size and torque-to-weight ratios of the hydrostatic-drive system, produce noisier pumps, motors, and hydrostatic controls. Also, although weight and size of the hydrostatic components can be reduced by higher operating speeds, increased speeds will reduce the system's efficiency.

PREDICTIONS

It is predicted that the efficiency of the hydrostatic-drive system will improve through continued R&D. Predicted technological advances include:

(a) Better thermostabilization of material to permit closer clearance tolerance.

(b) Stabilization of hydraulic fluid viscosity.

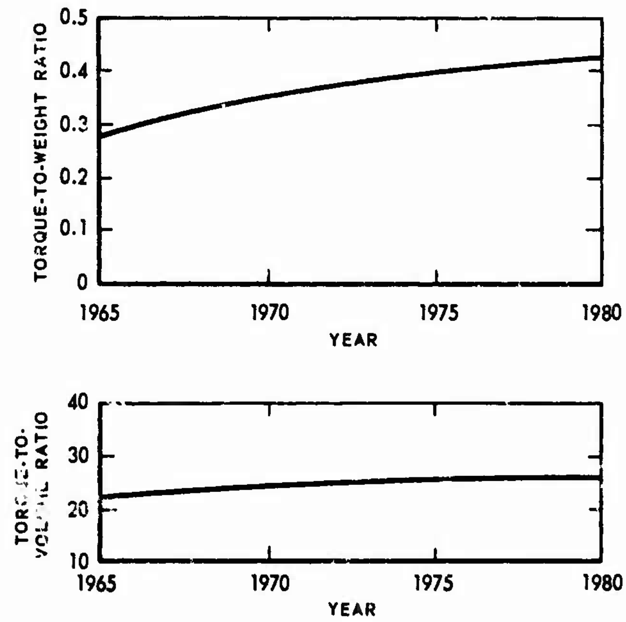

Fig. II-32—Predicted Specific Ratio Improvements for Hydrostatic Power Trains

(c) Improvement of lubricating qualities of hydraulic fluid.

(d) Reduction of noise levels at high operating pressures through use of improved valves and porting.

(e) Development of high-speed vane pumps with pivoted tips that will (1) increase operating pressures to 8000 psi at speeds up to 30,000 rpm, (2) permit direct coupling of the pump to a high-speed engine (such as the gas turbine), and (3) reduce both size and weight of the pump as well as of the system, since no gear reduction is required.

(f) Decrease in cost as production volume increases and production methods improve.

Predicted specific torque-to-weight and torque-to-volume ratio improvements for hydrostatic-drive power trains through 1980 are illustrated in Fig. II-32. Predicted efficiency improvements for hydrostatic-drive power trains through 1980 are illustrated in Fig. II-33.

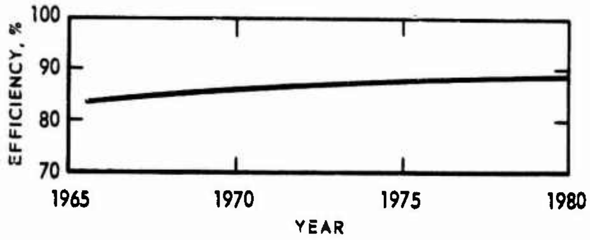

Fig. II-33—Predicted Improvement in the Efficiency of Hydrostatic-Drive Systems

CONCLUSIONS

Government R&D of hydrostatic-drive systems for application in tactical vehicles similar to those being developed for commercial purposes is not warranted, since industry will continue to develop these systems successfully.

However, industry is not developing hydrostatic transmissions or power trains suitable for special types of tactical vehicles that are, or may be, in the military system.

Therefore Government R&D is required on hydrostatic-drive systems for tactical vehicles where present mechanical or hydrokinetic transmissions or power trains cannot provide the desired physical and performance characteristics.

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Chapter 21

ELECTRIC DRIVES

INTRODUCTION

An electric drive is a power-conversion device designed to transmit power, through an electrical circuit, from the power source to the wheel or sprocket of a vehicle. There are two basic types of electric-drive systems: dc and ac. Both types are discussed in this chapter.

Electric-drive systems can use power from an ac or dc electrical power source. For example the power source may be an energy storage device or externally fed power lines. Also the electric-drive systems can convert mechanical power, obtained from energy-conversion devices such as a reciprocating or gas-turbine engine, to electrical energy. Later, at the point of application, the electrical energy is reconverted to mechanical energy.

In theory, electric drives are attractive for application in vehicles in which use of more conventional power-conversion devices is difficult or not feasible. Amphibious vehicles with retractable wheels, rear-wheel-power semitrailers and scrapers, multiple-unit articulated vehicles, and vehicles that require a low silhouette are examples of this type of vehicle.

USAERDL, Ft Belvoir, Va., and ATAC at Warren, Mich., were visited during the conduct of this study. The active responsibility for R&D of electric drives for tactical vehicles is held at these two military installations.

Information on electric drives also was obtained from publications and personnel in industry and Government agencies. Industrial plants visited included GMC Defense Research Laboratories, Santa Barbara, Calif.; AiResearch Manufacturing Company, Torrance, Calif.; General Atomics Division of General Dynamics, San Diego, Calif.; Solar Division of International Harvester Corporation, San Diego, Calif.; FMC Corporation, San Jose, Calif.; Lear Siegler Inc., Cleveland, Ohio; and the Bay Area Rapid Transit District Test Site at Concord, Calif.

Other agencies and industries that have been contacted include the Harry Diamond Laboratories, Washington, D. C.; Pima Mining Company, Tucson, Ariz.; and the LeTourneau Company, Longview, Tex.

DC ELECTRIC-DRIVE SYSTEM

Direct-current electric-drive systems are capable of using electrical energy obtained direct from batteries or unique energy-conversion devices or energy from a mechanical power source after such energy has been converted through a generator to direct current. Early versions of dc electric-drive systems were used for automobiles, trucks, streetcars, and locomotives. These systems were powered by batteries or electric power lines having an external feed. These electric-drive systems were reliable and economical to operate, but as technological advances were made to other types of energy and power-conversion devices the application of electric-drive systems changed. At present dc electric drives powered by batteries are found in golf carts, forklift trucks, delivery trucks, short-range automobiles. and other commercial vehicles where quiet operation or freedom from air pollution is required.

There are no known battery-powered tactical vehicles in the military system although many such commercial vehicles are used by the military in logistical operations.

Recently a battery-powered electric-drive system was installed in an M37 ¾-ton truck to demonstrate the capabilities of the vehicle. This vehicle had excellent acceleration and operated quietly, but the weight of the vehicle showed substantial increase, and most of the cargo space and capacity was used to house the vehicle batteries. A weight comparison of a mechanical drive system with two battery-powered dc electric-drive systems, all for an M151 vehicle, is made in Table II-3.

The potential of dc electric-drive systems, operating from "electrochemical energy-storage system" (EESS) devices, is being investigated both by industry and the military to determine the possible uses of these systems in vehicles. The EESS devices require fuel and accessories to constitute a complete system. Two prototype systems of those produced on an experimental basis were considered by RAC in this study. One system uses an air-zinc couple, and the other uses a molten electrolyte lithium-chlorine couple. These prototype systems have achieved an energy density of from 50 to 60 watt-hr/lb and from 215 to 225 watt-hr/lb, respectively, and are discussed in some detail in the section "Batteries." The weight of an EESS-powered electric-drive system for an M151 vehicle is compared with that of a mechanical drive system for the same vehicle in Table II-4.

On the basis of weight alone, neither a battery-powered electric-drive system nor an EESS device offer much promise for use in tactical vehicles unless quiet operation should be a prime requisite.

Dc electric-drive systems, with conventional reciprocating engines as energy-conversion devices, were installed in approximately 250 T23 medium tanks during WWII. Tank performance when using the electric drive was acceptable, but the system increased the weight of the vehicle over that of a tank having a mechanical drive. However, installation of the electric drive in these tanks provided the information necessary to establish some fundamental concepts for steering, dynamic braking, regenerative steering, and vehicle control.

A dc electric-drive system was installed in a 65-ton ore hauler powered by a reciprocating engine (Fig. II-34). The electric-drive system was installed to determine what advantages this system had to offer over the mechanical

TABLE II-3

Weight Comparison of Battery-Powered Dc Electric-Drive System with a Mechanical-Drive System[a]

Item	Electric-drive weight, lb		Mechanical-drive weight, lb
	Lead-acid battery	Silver-zinc battery	
Spark-ignition engine	—	—	440[b]
Gasoline	—	—	252[c]
Electric drive	720	720	—
Battery	2790[c,d]	840[c,e]	—
Total	3510	1560	692

[a]An 80 percent overall efficiency is assumed and systems sized for M151 vehicle.
[b]72 hp.
[c]Based on 10-hr day and 45-hp average demand.
[d]Power density 15 watt-hr/lb.
[e]Power density 50 watt-hr/lb.

TABLE II-4

Weight Comparison of EESS-Powered Electric-Drive System with a Mechanical-Drive System[a]

Item	Electric-drive weight, lb		Mechanical-drive weight, lb
	Zinc-air system	Molten electrolyte system	
Spark-ignition engine	—	—	440[b]
Gasoline	—	—	252[c]
Electric drive	720	720	—
EESS device	763[c,d]	186[c,e]	—
Total	1483	906	692

[a]An 80 percent overall efficiency is assumed and systems sized for M151 vehicle.
[b]72 hp.
[c]Based on 10-hr day and 45-hp average demand.
[d]Power density 55 watt-hr/lb.
[e]Power density 225 watt-hr/lb.

Fig. II-34—65-ton Ore Hauler with Dc Electric Drive

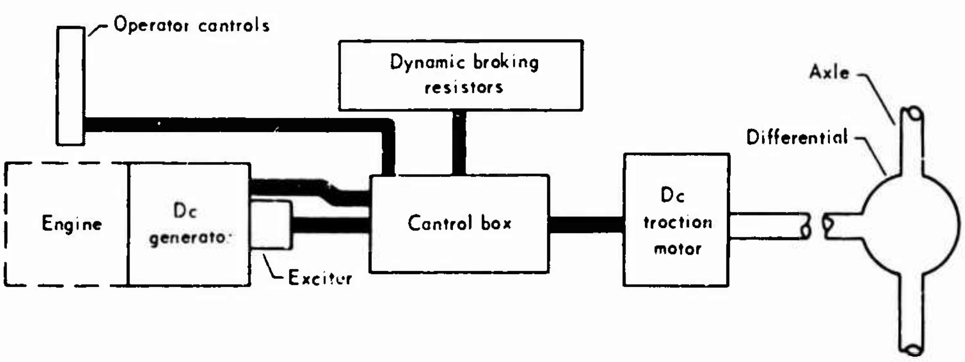


Fig. II-35—Dc Electric-Drive System with Common Traction Motor for All Drive Wheels transmission it replaced. A schematic diagram of the electric-drive system is shown as Fig. II-35. Although the vehicle did demonstrate better braking capability following the installation, other differences in performance were not noted. However, the net weight of the vehicle increased 10,000 lb; the overall vehicle cost increased 15 percent; and the electric-drive system required more space in the vehicle than the system it replaced. The efficiency of the electric-drive system as a function of road speed was determined and is plotted in Fig. II-36.

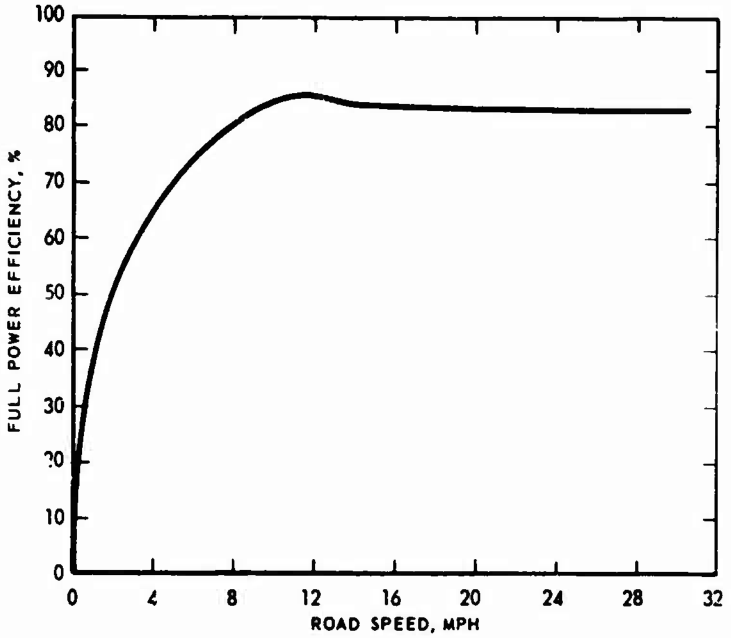
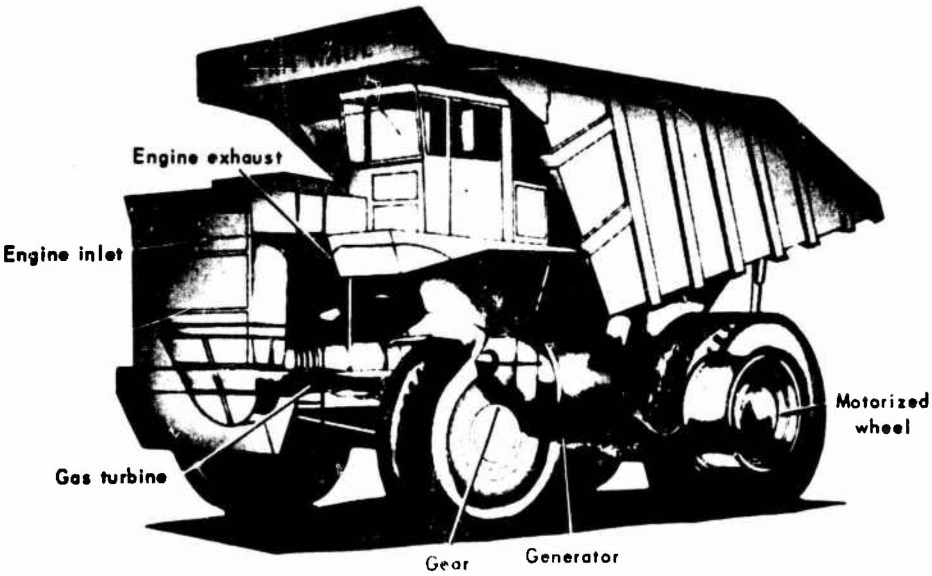


Fig. II-36—Dc Electric-Drive-System Efficiency as a Function of Road Speed of a 65-ton Ore Hauler

A dc electric-drive system incorporating individual-wheel drives was installed in a 100-ton ore hauler. The unit is illustrated in Fig. II-37. A compression-ignition engine driving a dc generator was used as the power source for this vehicle. The output of the generator was fed through controls to each individual-wheel drive motor shown in the cutaway view as Fig. II-38. In this application the weight of the dc electric-drive system was 30 lb/hp. This application proved successful since it permitted the designer to design the vehicle without regard to the restricting requirements of a mechanical drive line. The operator was able to handle the vehicle with ease since the controls were simple. As a result payload cycle time was reduced. The vehicle's reliability and maintainability were estimated to be improved over similar mechanical-drive system vehicles.

A recent dc electric-drive system was installed in a 100-ton ore hauler powered by a gas turbine (Fig. II-39). The gas-turbine engine powered a dc generator, through a set of speed-reduction gears, and electric power from



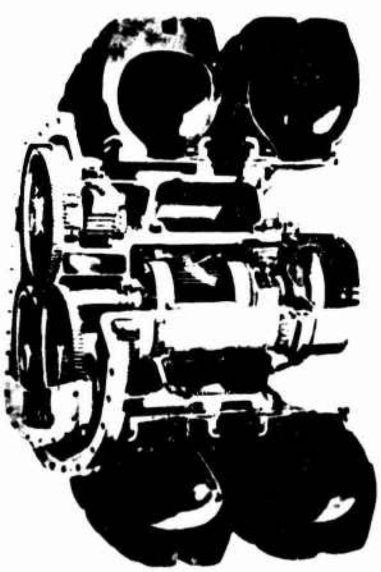


Fig. II-38—Dc Electric-Wheel Drive Motor the generator was distributed to each wheel motor. Tests then revealed that the generator was overloading the gas turbine because the gearbox reduction ratio was too high. This deficiency is now being corrected; the gearbox reduction is being decreased to increase the generator speed. The modified vehicle test results will not be obtained in sufficient time to be included in this study.

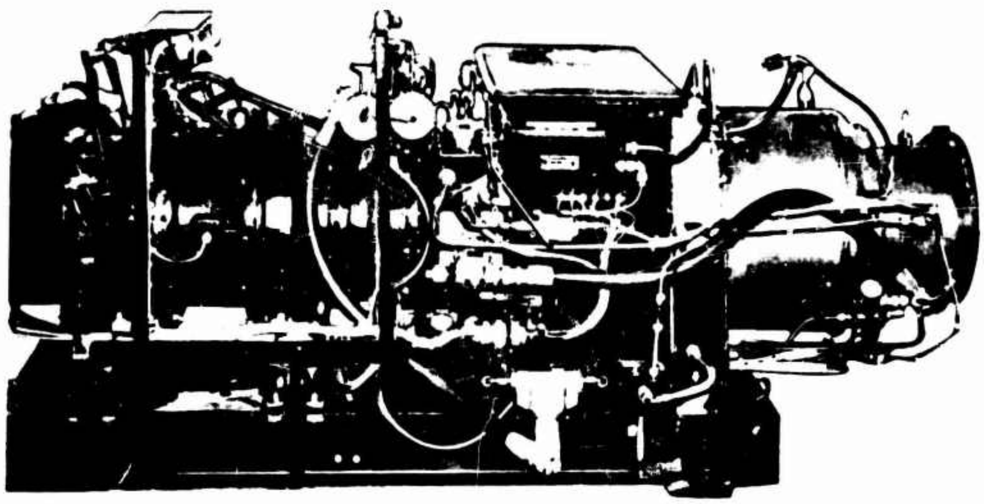


Fig. II-39—Gas-Turbine Engine with a Dc Generator

A dc electric-drive system, powered by a gas turbine, was installed by the US Army in an Arctic Overland Train in which all wheels were powered (Fig. II-40). This type of vehicle is excellent for application of an electric-drive system since the engineer designing the unit has freedom in locating the power-conversion components to obtain optimum positioning with dynamic braking at all wheels. The electric-drive system has given technical feasibility to the Overland Train.

In those test installations where a dc electric-drive system has been coupled to a conventional engine, indications have been that the systems then have the potential of improving tactical vehicle capability. With this system the designer has more latitude in positioning the various vehicle components. As a result drive motors can be placed at each wheel to increase vehicle mobility, desired vehicle configuration can be more readily achieved, and optimum wheel loading can be obtained through a better distribution of vehicle weight. However, the dc electric-drive system does increase the cost, size, and weight of a vehicle over that of one incorporating a mechanical power-conversion device.

In consideration of the foregoing it appears that dc electric-drive systems have applications only for special tactical vehicles that have a requirement for physical and performance characteristics that cannot be met through use of the conventional mechanical-drive systems.

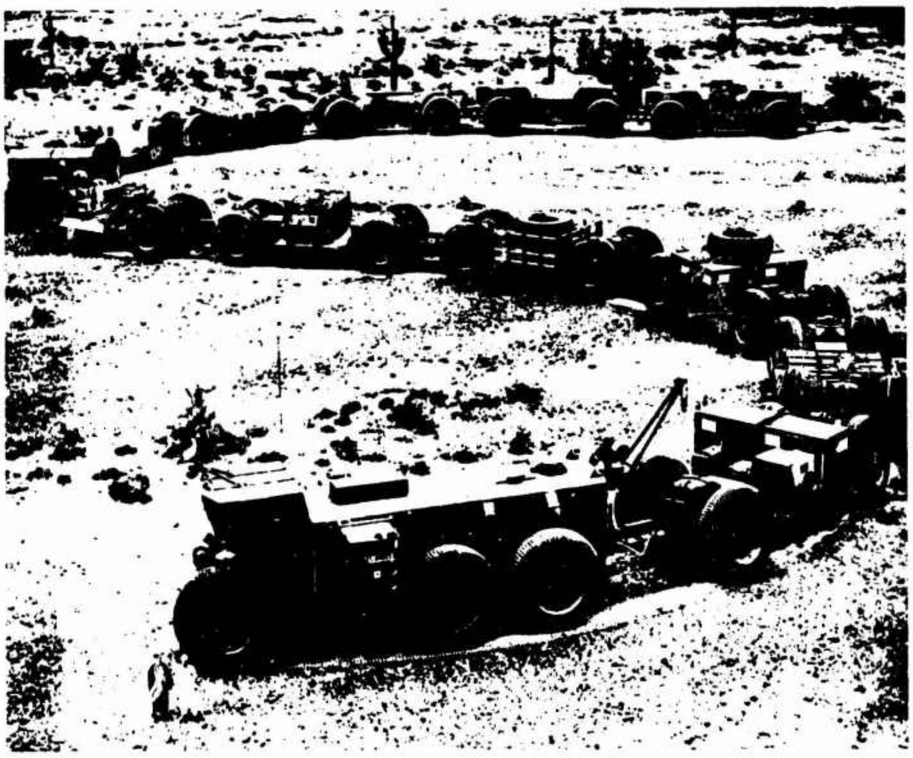


Fig. II-40—Overall View of Overland Train Mark II

ALTERNATING-CURRENT ELECTRIC-DRIVE SYSTEM

The ac electric-drive system operates from an ac source, usually generated by rotating equipment. Batteries, electromechanical storage systems, and fuel cells can also be converted to alternating current, but additional equipment is required to convert the direct current produced. This additional equipment would add weight to, and increase the size of, the system. Therefore only ac electric-drive systems powered by rotating equipment are considered in this portion of the study.

Ac generators and motors have had various and widespread uses for every conceivable type of application throughout the world. These motors normally operate from constant voltage and frequency sources but in most instances are not suitable for application in tactical vehicles since they do not develop sufficient torque at both low speeds and highway speeds. By varying the frequency and voltage the early ac motors could produce high torque at low speeds, but this required additional equipment that added considerable weight and proportionately increased the size of the system.

The ac induction motor was considered for use in an electric-drive system because it is brushless, smaller, and lighter in weight than a dc motor counterpart. The motor windings of an ac induction motor are in the stator only, which makes the motor easier to cool.

The recent development of silicon-controlled rectifiers (SCR) has made possible the design of control systems that greatly reduce the size and weight of those components that control the frequency and voltage of energy produced by a generator.

The ac variable-frequency electric-drive system was conceived in 1959, and a design study then was initiated by the US Army. In 1961 the results of the study showed promise sufficient to warrant component development. Concurrently a study for an electric-drive system having applicability to track-laying vehicles was initiated. This second study, which compared 17 different electric-drive power-train systems, was completed in 1962. After the size, weight, and efficiency of the units considered in this study were evaluated, an ac variable-frequency electric-drive system was determined to hold the most promise for application in the near future. The system envisioned was predicted to operate at high speed, to have good efficiency, and to use brushless alternators that could be coupled to a conventional reciprocating engine. Recognition was given the fact that for large vehicle applications this type of system could readily be coupled to a gas-turbine engine.

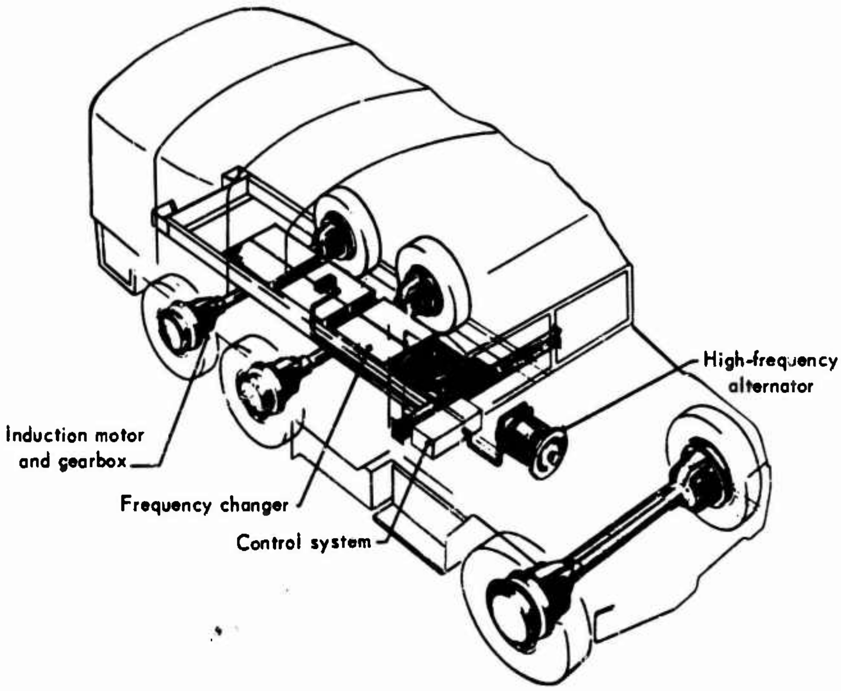


Fig. II-41—Ac Electric-Wheel-Drive Component Installation

An ac variable-frequency electric-drive system was installed in an M34 $2\frac{1}{2}$-ton 6x6 truck (Fig. II-41). A 6x6 truck was selected for the installation since this vehicle is used extensively by the military and was readily available. A block diagram of the system installation is shown in Fig. II-42. The drive consists of an alternator mounted to an engine, a signal generator and exciter, a frequency changer, system controls, and individual-wheel drive motors. The main purpose of this test-bed installation was to apply the technology of the cycloconverter (variable-frequency controller) in an electric-drive system and to determine the feasibility of powering all wheels in a <u>wheeled</u> vehicle. The

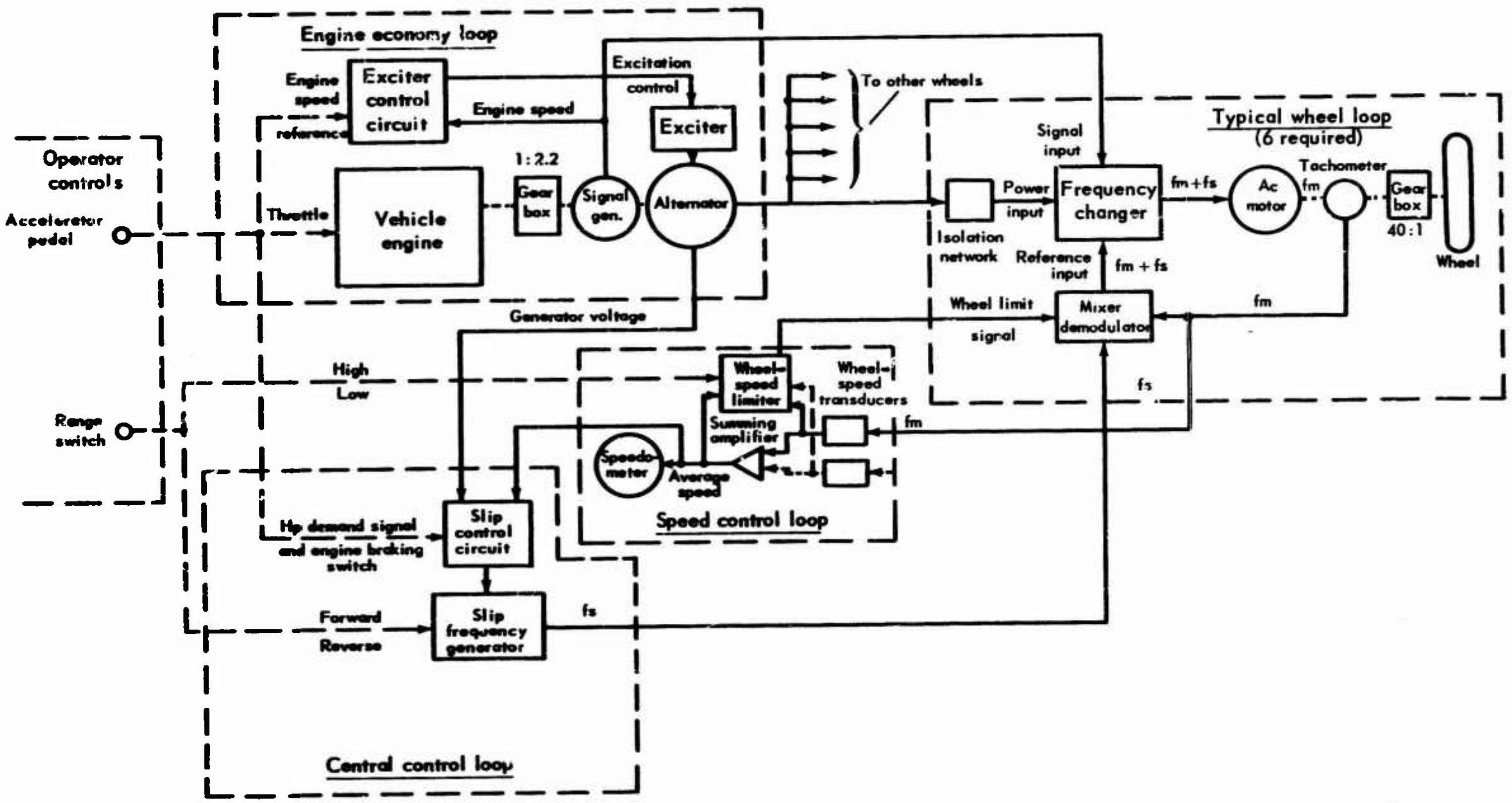


Fig. II-42—Block Diagram of Ac Variable-Frequency Electric-Wheel-Drive System vehicle was operated for several thousand miles over highways and various types of cross-country terrain. The results of these operations indicated the feasibility of using ac electric drives in tactical vehicles. However, this type of system increased the weight and lowered the operating efficiency of the vehicle below that experienced with a mechanical drive. In addition the vehicle could not obtain top highway speed or negotiate a 60 percent slope at the required speed. The weight of the test-bed vehicle could have been less, but to reduce costs many commercial and readily available electric components were used. In addition, components were located in the vehicle to be readily accessible to test personnel.

TABLE II-5

Weight and Volume Comparison of the Ac Electric Drive and the Mechanical Drive in an M34 2½-ton Truck

Component	No.	Weight each, lb	Size each	Total weight, lb	Total volume, ft^3
			Electric Drive		
Speed-increasing gearbox (1.0–2.2)	1	40[a]	1.5 ft^3 [a]	40	1.50
Alternator	1	404	26¼ in. × 16-in. diameter	404	3.06
Frequency changers	6	100	8.75 × 25.25 × 11.71 in.	600	9.00
Induction drive-wheel motors	6	167	10.8-in. diameter × 11.6-in. high	1002	3.70
Speed-reducing gearbox (40:1)	6	250[a]	12-in. diameter × 12 in.[a]	1500	4.72
Wheel cross shafts	3	20[a]	4-in. diameter × 56 in.[a]	60	1.22
Control set	1	40[a]	1.0 ft^3 [a]	40	1.00
Yoke	6	75[a]	0.166 ft^3 [a]	450	1.00
Total				4096	25.20
			Mechanical Drive		
Transmission	1	300	5.5 ft^3 [a]	300	5.50
Axles and differentials	3	580[a]	3 ft^3 [a]	1740	9.00
Drive shafts	2 1	20[a] 16[a]	1.2 ft^3 [a]	56	1.20
Brake and wheel hub	6	8[a]	0.5 ft^3 [a]	48	3.00
Total				2144	18.70

[a]Estimated.

The estimated weight and volume of the ac variable-frequency electric-drive system is compared with that of the mechanical drive in Table II-5. The best operating efficiency of the electric-drive system was 84 percent. The mechanical drive proved to have an efficiency of 89 percent.

Additional testing of this vehicle is planned to determine the capability of the electric-drive system with regard to the following items:

(a) Component durability and life
(b) Performance and reliability
(c) Braking grid effectiveness for dynamic braking
(d) Effectiveness of the redesigned alternator with rotary exciter
(e) Effectiveness of a smaller and lighter frequency changer

A new type of electric-drive system incorporating a dc link is now being installed in an M113 tracked vehicle (Fig. II-43). Tests will be made to determine the feasibility of using this system for tracked vehicles. This test-bed installation was made to obtain performance data for comparison with data obtained from a system having a mechanical power train, and to obtain quantitative data to determine what modifications may be required in the future to permit use of an electric-drive system.

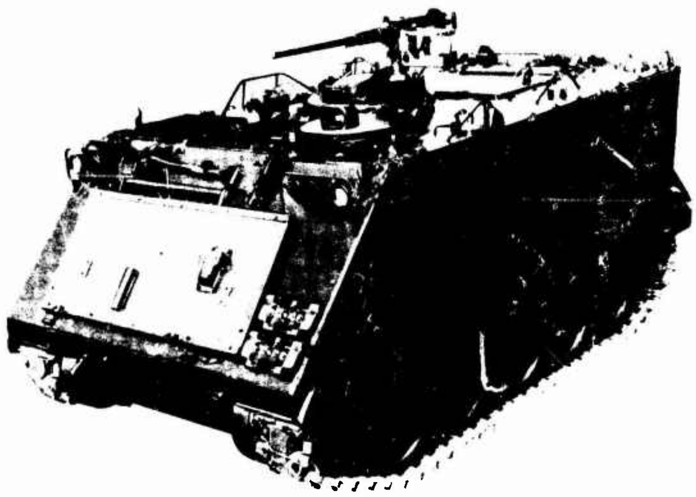


Fig. II-43—M113 Test-Bed Vehicle with Ac Electric-Drive System

Standard generators may be used in the system with the dc link, or the system can be adapted to use dc power sources such as batteries or fuel cells. A block diagram of this system is shown in Fig. II-44. The electric-drive system consists of an alternator (mounted to the engine), a rectifier and inverters, system controls, and two traction motors. For ease of installation the induction motors were affixed to each other and connected to the final drives (at each sprocket).

The performance of this test-bed vehicle, i.e., horsepower as a function of road speed, is illustrated in Fig. II-45. The prediction is made that the electric-drive system will have the capability of operating at essentially constant horsepower over most of the speed range indicated. The weight and volume of the ac electric-drive system, as compared with the mechanical drive it replaced, is shown in Table II-6. Note that the weight and volume required for the electric-drive system is considerably more than that required by the mechanical drive. However, it is predicted that the redesign permitted by technological advances and the incorporation of the latest state-of-the-art components will result in a substantial reduction of these values for the electrical-drive system.

A test-bed installation of an ac electric-drive system in a BEST earthmoving vehicle is now being made. An artist's concept of this vehicle is shown in Fig. II-46. Dynamotor testing of BEST is scheduled for the latter part of 1966. This vehicle is appropriate for application of an electric-drive system since the chassis is articulated, yet all wheels can be readily powered. A block diagram of this system is shown in Fig. II-47.

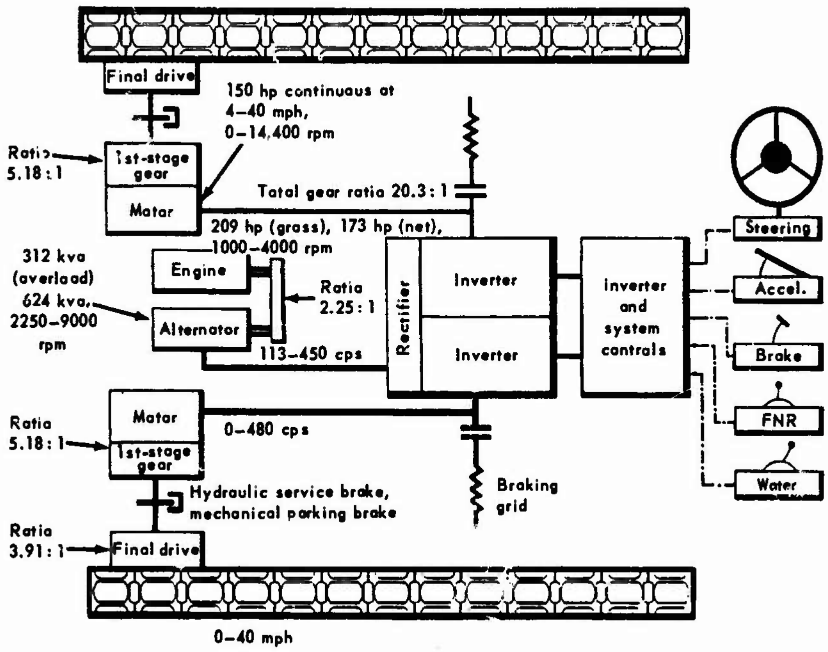


Fig. II-44—Diagram of Ac Electric-Drive System

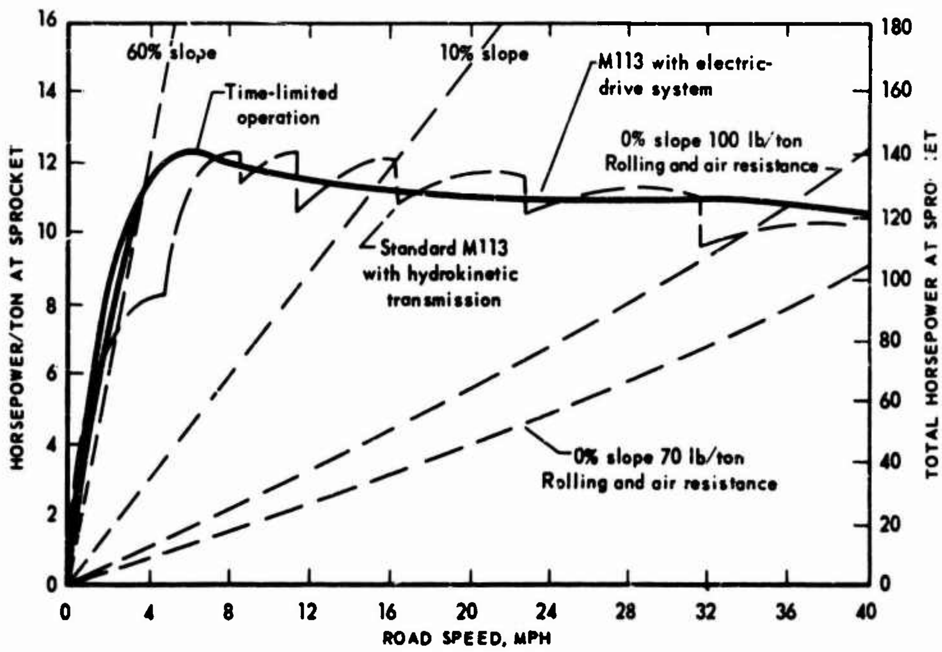


Fig. II-45—Vehicle Horsepower as a Function of Road Speed

TABLE II-6

Weight and Volume Comparison of the Ac Electric Drive and the Mechanical Drive in an M113 Test-Bed Vehicle

Component	No.	Weight each, lb	Size each	Total weight, lb	Total volume, ft³
			Electric Drive		
Chain drive and guard, step up (2.25 : 1)	1	120	8 × 3 × 24 in.[a]	120	0.30
Alternator	1	395	27.9 in. × 13.4-in. diameter	395	2.30
Rectifier and inverters	1 unit	1600	2[illegible] × 30 × 72 in.	1600	27.50
Controls	1	90	.5 ft³ [a]	90	0.50
Traction motors and gearbox (5.18 : 1)	1 unit	1200	8 ft³	1200	8.00
Braking grids	1 set	150	12 × 16 × 20 in.[a]	150	0.22
Total				3555	38.82
			Mechanical Drive		
Transmission TX 200-2A	1	450	5.8 ft³	450	5.80
Controlled differential FMC DS 200	1	508	9.5 ft³	508	9.50
Transfer case	1	150[a]	1.5 ft³ [a]	150	1.50
Drive shafts	2	20[a]	1.0 ft³ [a]	40	1.00
Total				1148	17.80

[a]Estimated.

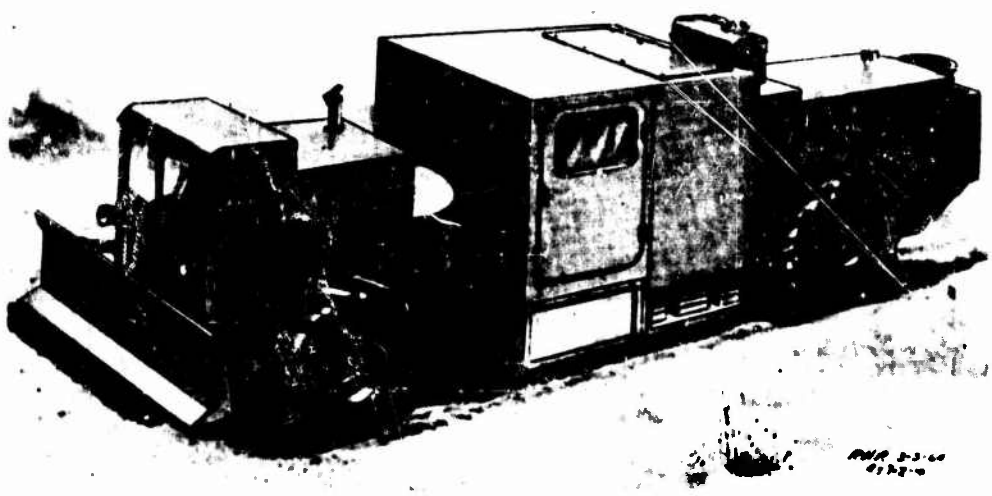


Fig. II-46—Ac Electric-Drive BEST Test-Bed Vehicle

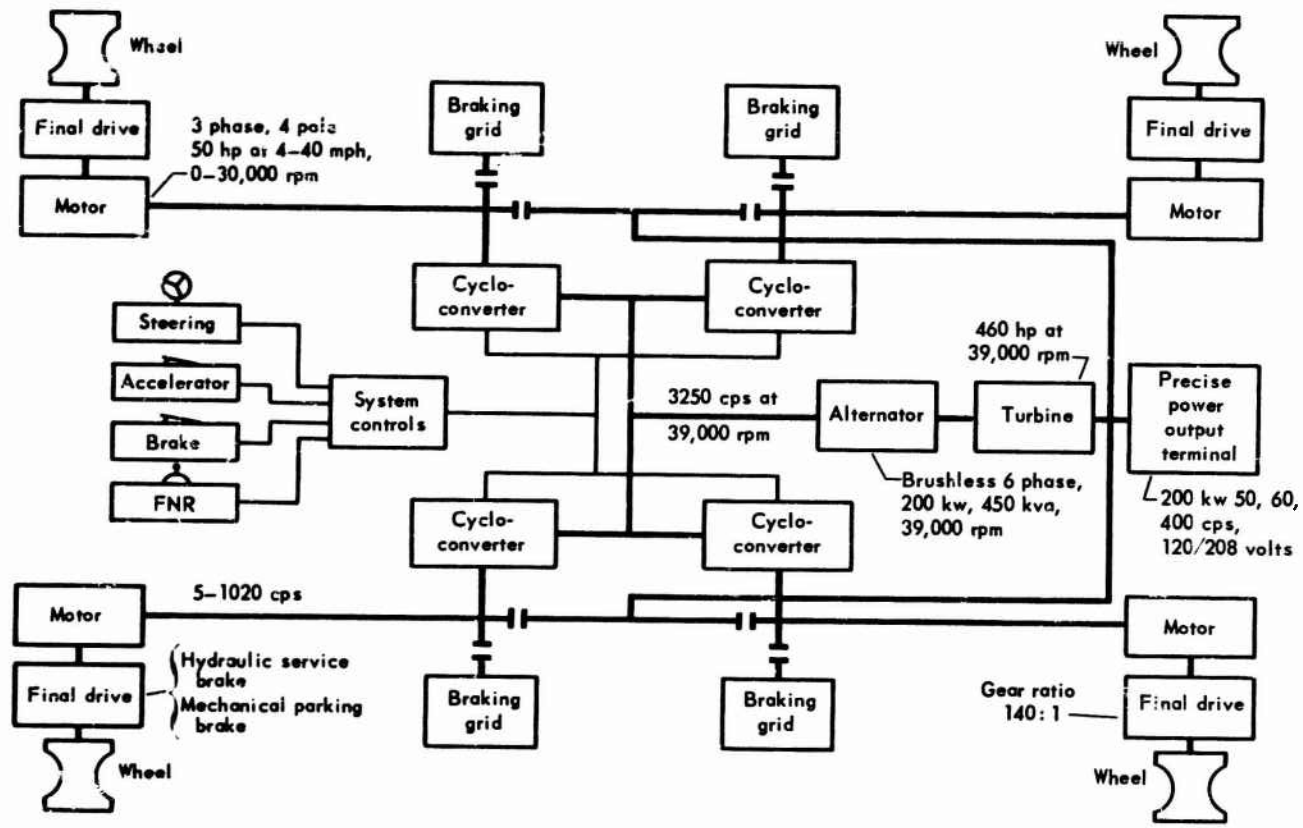


Fig. II-47—Schematic Diagram of Ac Electric-Drive System

The power source for BEST is a single-shaft gas-turbine engine coupled to an alternator (Fig. II-48). The alternator operates at the shaft speed of the gas turbine. High-frequency (3200 cps) power is fed to solid-state cycloconverters, where the voltage and frequency are controlled, and fed to the motors to produce the torque speed selected by the operator. The high-speed ac induction motors are capable of operating at constant horsepower over a wide speed

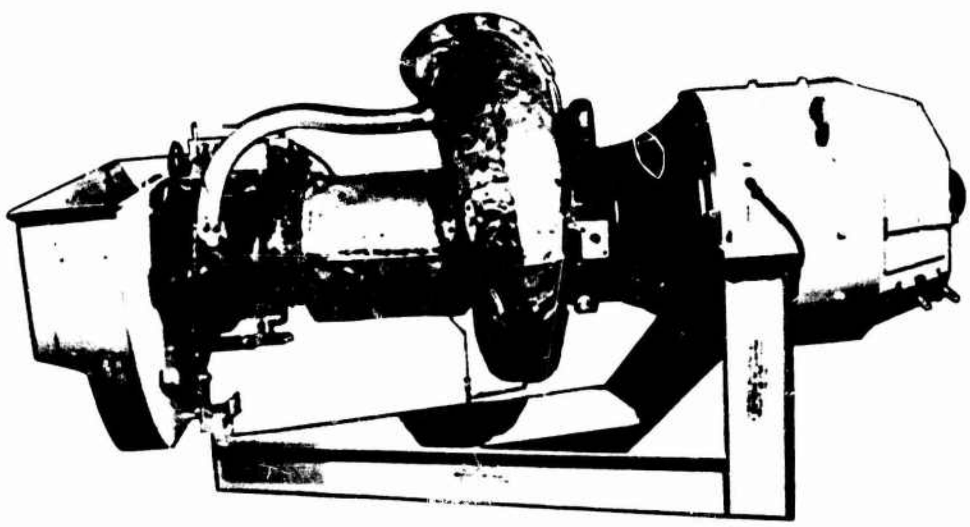


Fig. II-48—Gas Turbine with Ac Alternator

range. A final reduction gear is used between the motors and the wheels to increase torque and reduce the speed of the wheels. The performance characteristics of the vehicle were calculated, and the horsepower output as a function of road speed is illustrated in Fig. II-49. The efficiency of the overall electric-drive system as a function of road speed is illustrated in Fig. II-50.

The weight and size (volume) of the ac electric-drive system and the mechanical transmission (incorporating a hydrostatic front-wheel-assist system) that was replaced by the electric-drive system in the BEST vehicle are presented in Table II-7. The electric-drive system was found to be considerably heavier and larger than the mechanical system with the hydrostatic wheel-drive assist.

At the time of writing, an ac electric-drive system with a brushless synchronous motor was being installed in an M35 2½-ton 6×6 military truck. The electric wheel motors exhibit high torque characteristics similar to those of a dc motor, and therefore the system is often referred to as a "dc brushless motor drive system." An artist's concept of such a system in a vehicle is shown as Fig. II-51. A block diagram of this electric-drive system is shown as Fig. II-52. The weight and size (volume) of the ac electric-drive system is compared with that of a mechanical-drive system in Table II-8. The weight and size of the ac electric-drive system is shown to be considerably greater than that of the mechanical-drive system.

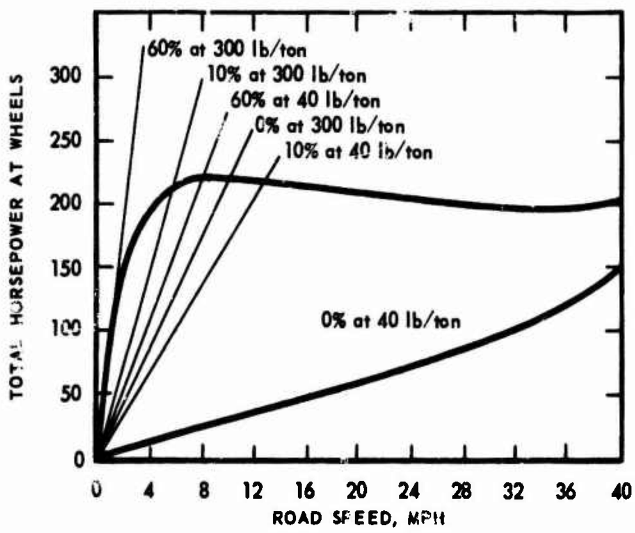


Fig. II-49—BEST Vehicle Horsepower as a Function of Road Speed

Rolling resistance = 40 lb/ton (paved), 300 lb/ton (rough terrain).

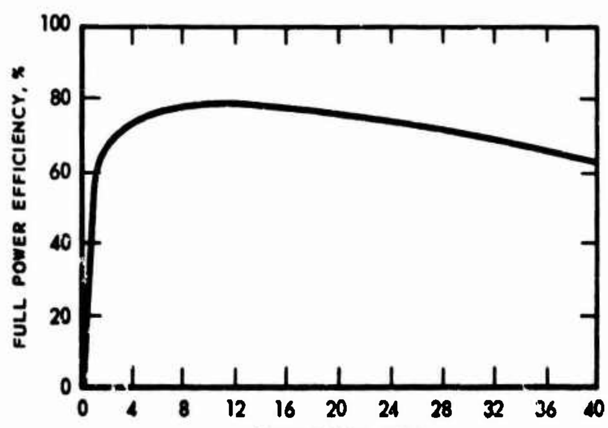


Fig. II-50—Overall Electric-Drive System Efficiency as a Function of Vehicle Road Speed

TABLE II-7

Weight and Volume Comparison of the Ac Electric Drive and the Mechanical Drive in a BEST Vehicle

Component	No.	Weight each, lb	Size each	Total weight, lb	Total volume, ft^3
			Electric Drive		
System controls	1	50 (estimate)	0.6 ft^3	50	0.60
Alternator including exciter	1	780	30 × 16.5-in. diameter	780	3.70
Motor	4	300	15.5 × 16.5-in. diameter	1200	7.73
Converters	4	250	15 × 15 × 36 in.	1000	18.79
Final reduction gearbox[a] (139.5 : 1)	4	1000	23 × 22.5-in. diameter	4000	9.40
Total				7030	40.22
			Mechanical Drive		
Transmission	1	1186	11.5 ft^3	1186	11.50
Differential and housing	1	500	13.5 ft^3	500	13.50
Final drive	2	180	0.5 ft^3	360	1.00
Hydrostatic front wheel assist:					
Pump	1	140	0.52 ft^3	140	0.52
Motors	2	53	0.115 ft^3	106	0.23
Gearbox, reduction	2	190	2.3 ft^3	380	4.60
Control valves	4	20	8 × 4 × 3 in.	80	0.22
Reservoir, including 30 gal of oil	1	240	10 × 30 × 24 in.	240	4.17
Total				2992	35.74

[a]Includes final-drive reduction and brakes.

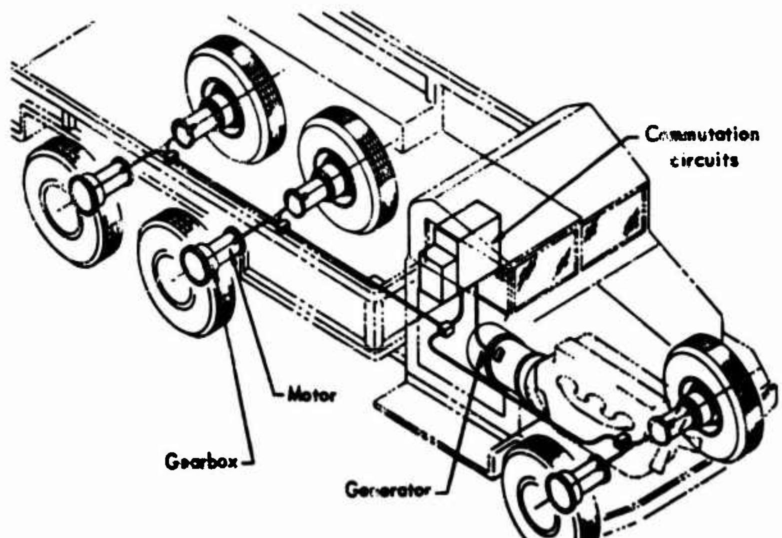


Fig. II-51—Ac Electric Drive (Dc Brushless Motor) in an M35 Test-Bed Vehicle

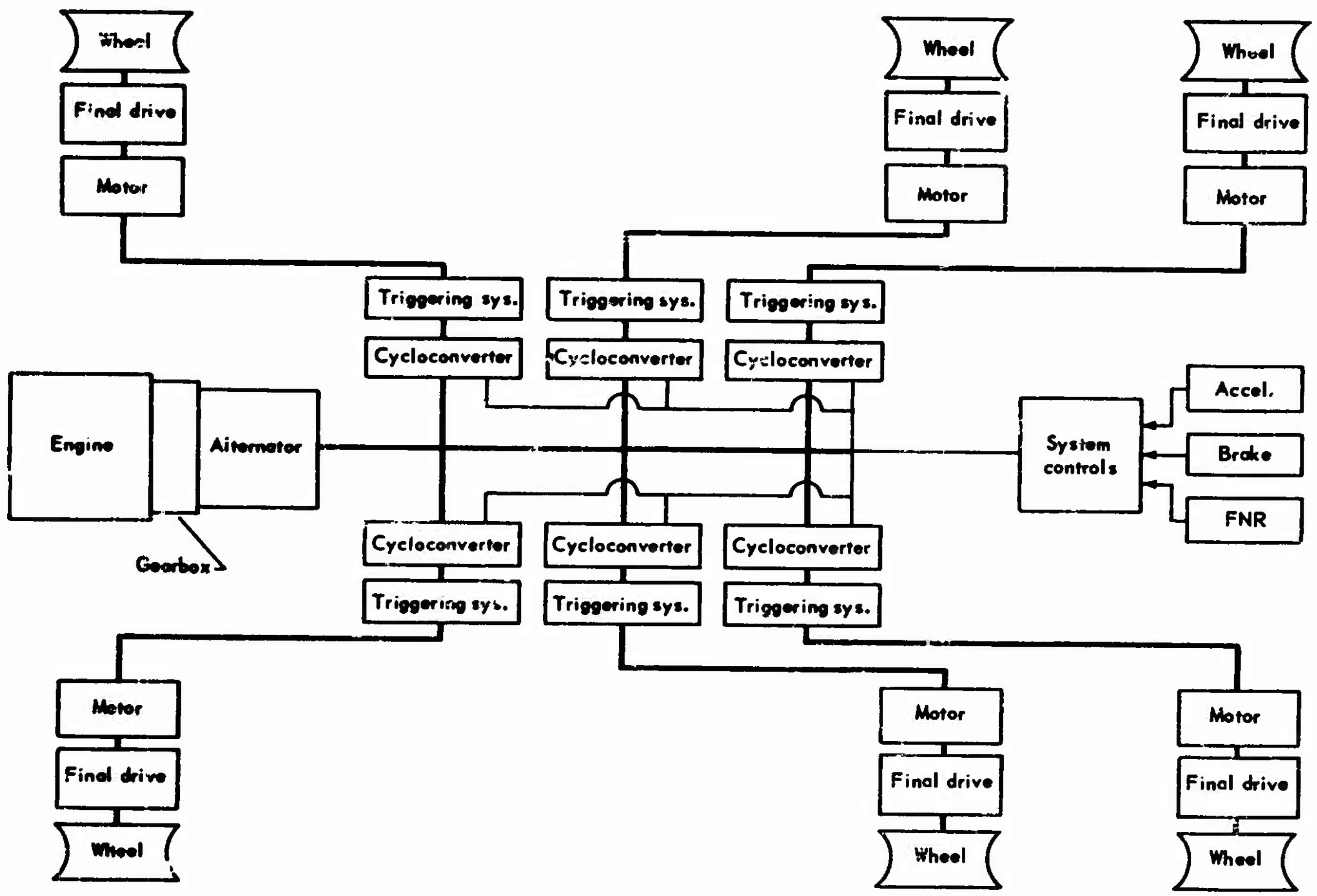


Fig. II-52—Schematic Diagram of Ac Electric Drive (Dc Brushless Motor)

The synchronous motor has operated satisfactorily on test stands over the complete speed and torque range required for those military vehicles in which it could be used. Therefore it is anticipated that the system should be successful when put to use in the actual operation of these vehicles.

TABLE II-8

Weight and Volume Comparison of the Ac Electric Drive and the Mechanical Drive in an M35 2½-ton Truck

Component	No.	Weight each, lb	Size each	Total weight, lb	Total volume, ft^3
			Electric Drive		
Alternator	1	600	20 × 16.5-in. diameter	600	2.48
Gearbox (speed-up)	1	150	9.5 × 20-in. diameter	150	1.73
Inverters	6	137	7 × 16 × 8 in.	137	3.12
Triggering system	6		6 × 19.63 × 8.5 in.		3.47
Motor	6	157	12 × 12-in. diameter	942	4.72
Motor gearbox	6	500	20 × 19-in. diameter	3000	19.70
System control	1	70	8 × 5 × 5 in.[a]	70	0.12
Wheel cross shaft	3	20[a]	56 × 4-in. diameter	60	1.22
Total				4959	36.56
			Mechanical Drive		
Transmission	1	300	5.5 ft^3	300	5.50
Axles and differentials	3	600	3 ft^3	1800	9.90
Drive shafts	2	20[a]			
	1	16[a]	1.2 ft^3	56	1.20
Brake and wheel hub	6	8[a]	0.5 ft^3	48	3.00
Total				2204	18.70

[a] Estimated.

EVALUATION

The various vehicle installations of the electric-drive systems discussed have provided data sufficient to indicate definite advantages of a dc electric-drive system over an ac electric-drive system, as well as some disadvantages. The prime advantage of the dc system lies in its simplicity as compared with the more complex ac system. A block diagram of the dc system is shown as Fig. II-53. A block diagram of the more complex ac system is shown as Fig. II-54. The ac system is more complex because of the controls required to vary the system frequency and voltage.

The efficiency of the dc electric-drive system is higher for most speed ranges than is that of the ac system. The efficiency of these two system types is compared in Fig. II-55. Further, the initial cost of a dc system is lower than that of an ac system owing primarily to the lesser complexity of the dc system.

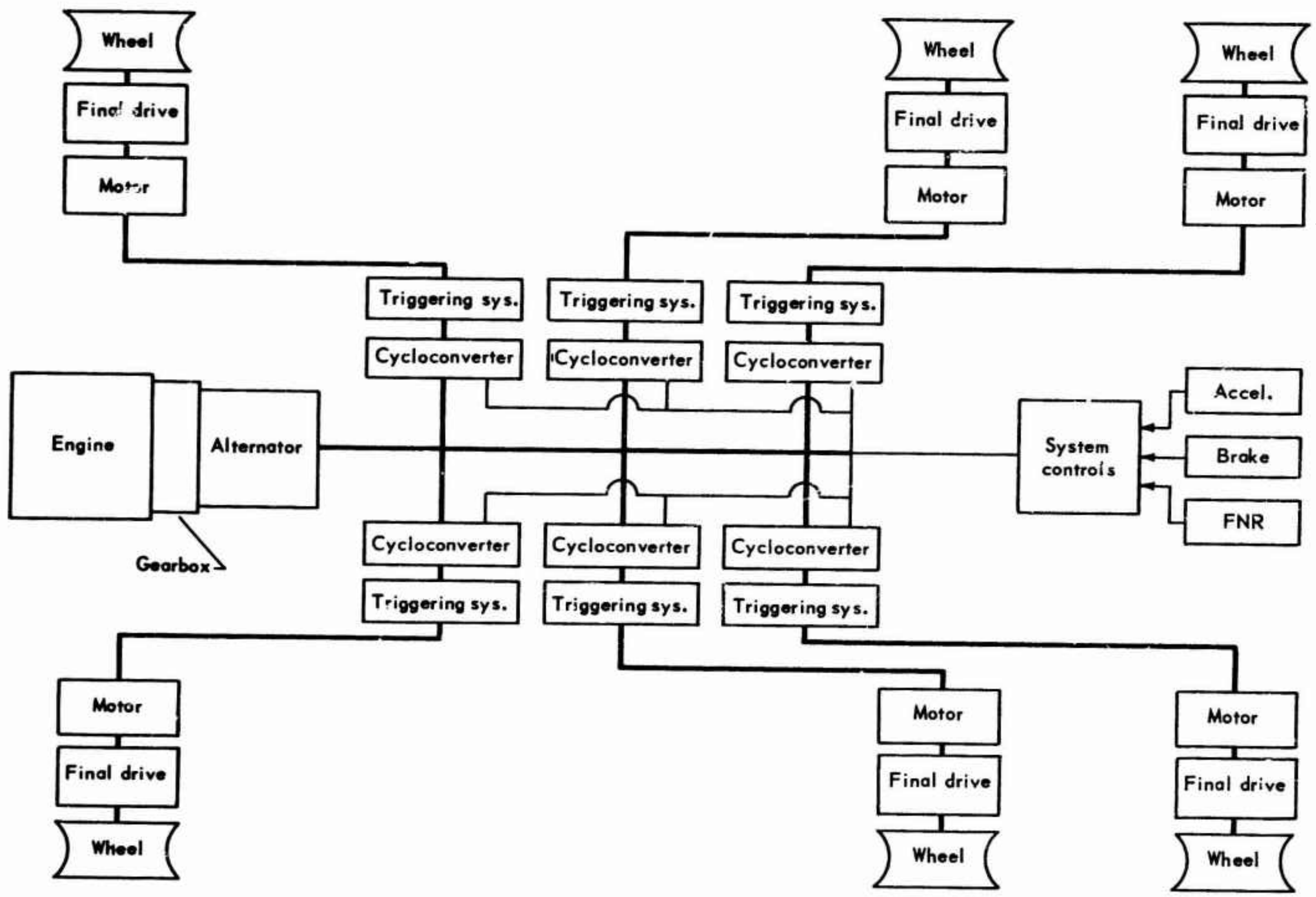


Fig. II-52—Schematic Diagram of Ac Electric Drive (Dc Brushless Motor)

The prime disadvantages of a dc electric-drive system are the weight and size (volume), both of which are greater than in an ac system. Motors and generators require brushes and commutators, which increase the weight and size of the dc system. Further, the motors require rotating windings, which limit speed and directly affect horsepower output. Another disadvantage of a dc system is that the motors and generators are difficult to waterproof and radio-suppress, due to commutation, more so than in an ac system.

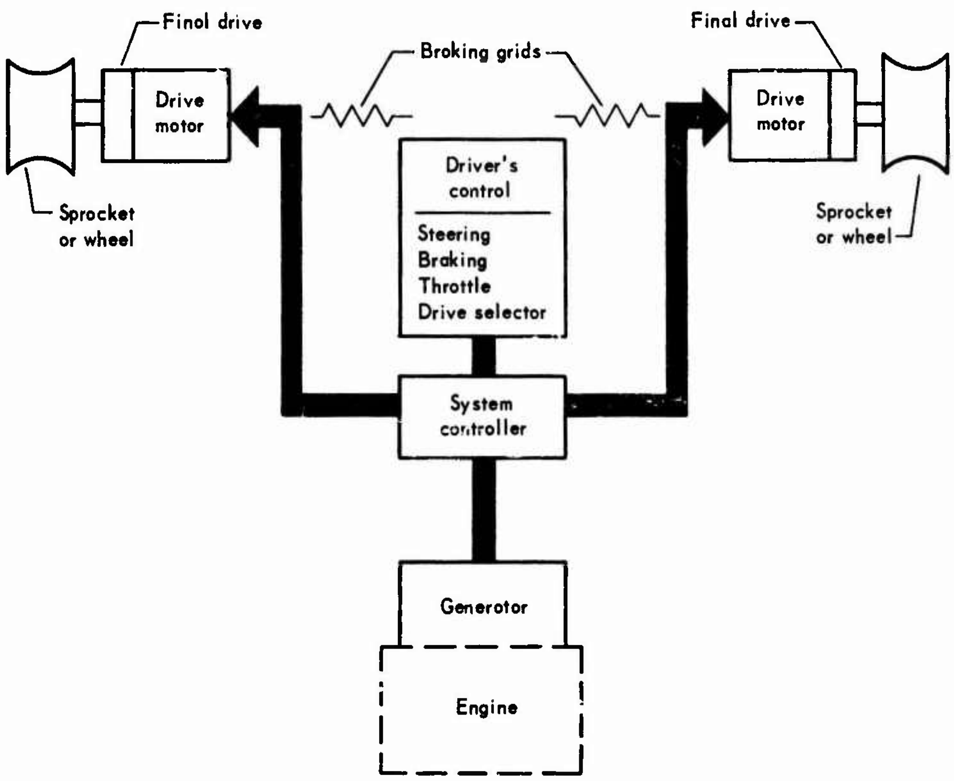


Fig. II-53—Dc Electric-Drive System with Individual Drive Motors at Each Wheel

Test-bed vehicles incorporating an ac electric-drive system provided the designer with test data on a variety of ac electric-drive systems, and with an evaluation of various ac electric-drive systems of small horsepower before adaptation of these systems to systems requiring greater horsepower. In addition electrical hardware not available in the commercial market evolved.

There has been large variation in the weight-per-horsepower and horsepower-per-volume specific ratings of ac electric-drive systems between tracked and wheeled vehicle installations. Also the specific weight and size of the system varied with the overall vehicle horsepower requirements. This is illustrated in Fig. II-56. It is apparent that, as the vehicle and horsepower size increase, a substantial improvement in an electric-drive-system specific rating is realized.

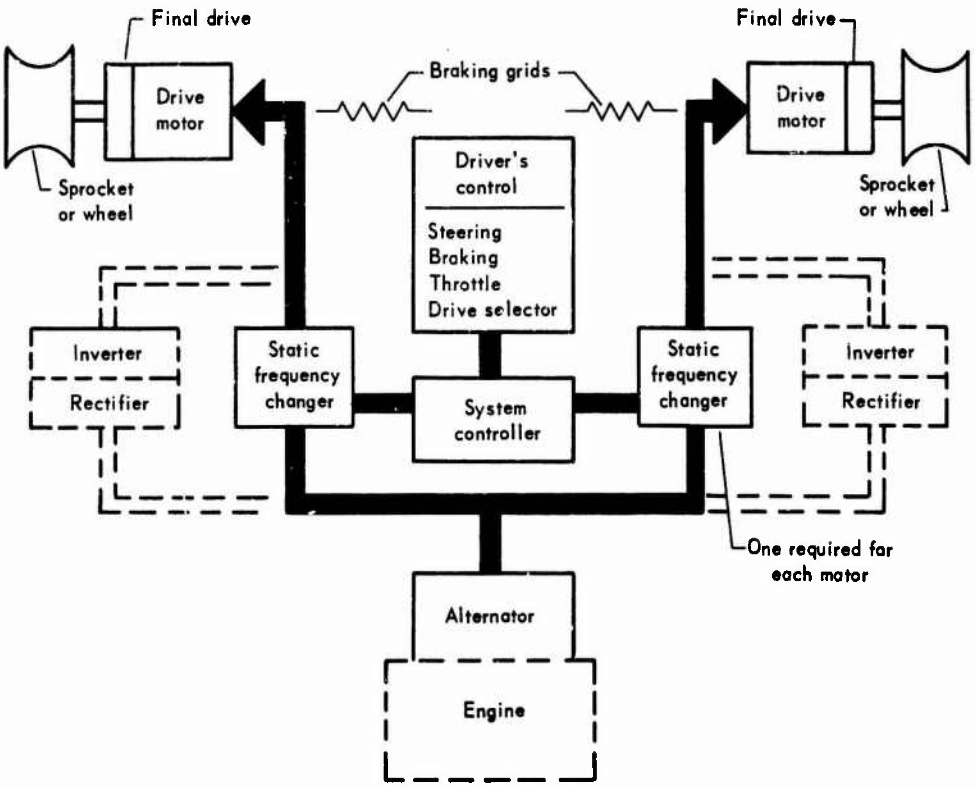


Fig. II-54—Ac Electric-Drive System

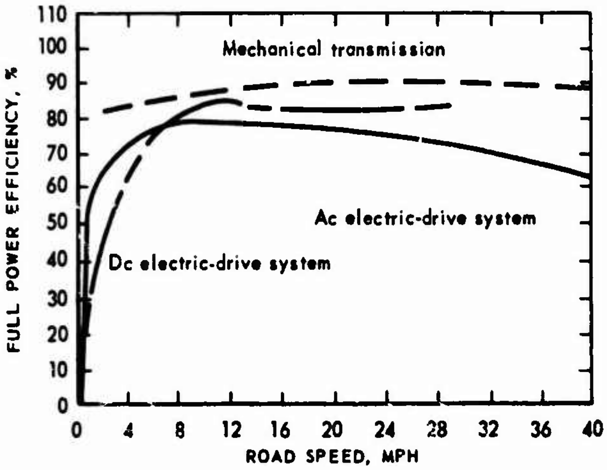


Fig. II-55—Efficiency Comparison of Various Drive Systems for Wheeled Tactical Vehicles

If the electric motors and generators are operated at higher speeds, voltages, and frequencies, the weight and size of the electric-drive systems can be reduced. In addition the use of lighter materials (primarily for housings) can make a contribution to system weight reduction.

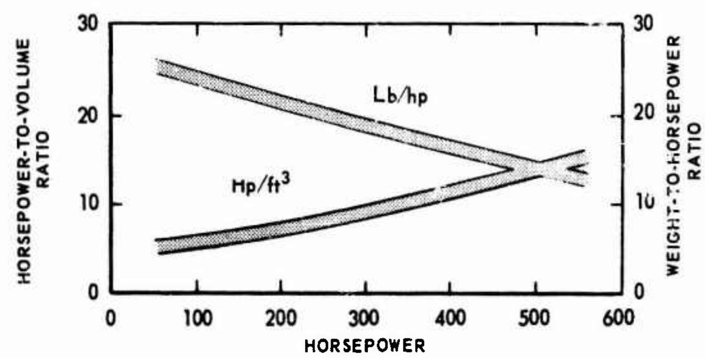


Fig. II-56—Weight and Volume Trend of Ac Electric-Drive Systems for Wheeled Vehicles as a Function of Horsepower Rating*

*Engine gross horsepower.

Alternators have been operated at 39,000 rpm. A speed this high permits direct alternator coupling to a gas-turbine engine. A realization of this possibility would further reduce vehicle weight since reduction gears would not be required. However, to achieve a high operating speed the use of a homopolar alternator containing a solid rotor is required. This type of alternator has an efficiency less than that of the slower-speed alternators that use a wound rotor. The permissible speeds of wound rotors, compared to speeds of solid rotors, in relation to the respective rotor diameters, are illustrated in Fig. II-57. The practical limit of a wound-rotor alternator (with the rotor having an approximate 8-in. diameter) is approximately 15,000 rpm whereas the practical limit of an alternator having a solid rotor of the same diameter is approximately 40,000 rpm. The higher operating speeds of a homopolar alternator readily achieve high frequencies, but the high speeds impose severe starting loads (when the alternator is coupled to a gas-turbine engine). Further, high-speed rotors require improved bearings and lubricants.

The use of lighter materials to reduce system weight is limited. Lighter materials can be used only for those components that are not required to carry high current or high flux density. The lighter material, then, is limited to structural components and would not appreciably reduce system weight.

Better cooling of the electrical components through improved air or oil-cooling techniques, or through the use of cryogenics, will permit electrical component operation at higher current densities. This would result in a decrease in system weight and size while offering comparable horsepower units. However, the use of cryogenics requires auxiliary equipment that adds weight and bulk to the system.

A simplification of control circuits, improved speed-control ratios, and microminiaturization could produce a large reduction in converter weight and volume. The greatest weight and size (volume) reduction in the electric-drive system will be realized in this area of controls.

An electric-drive system, be it ac or dc, offers some advantages and disadvantages for tactical vehicles, when compared with a mechanical-drive system.

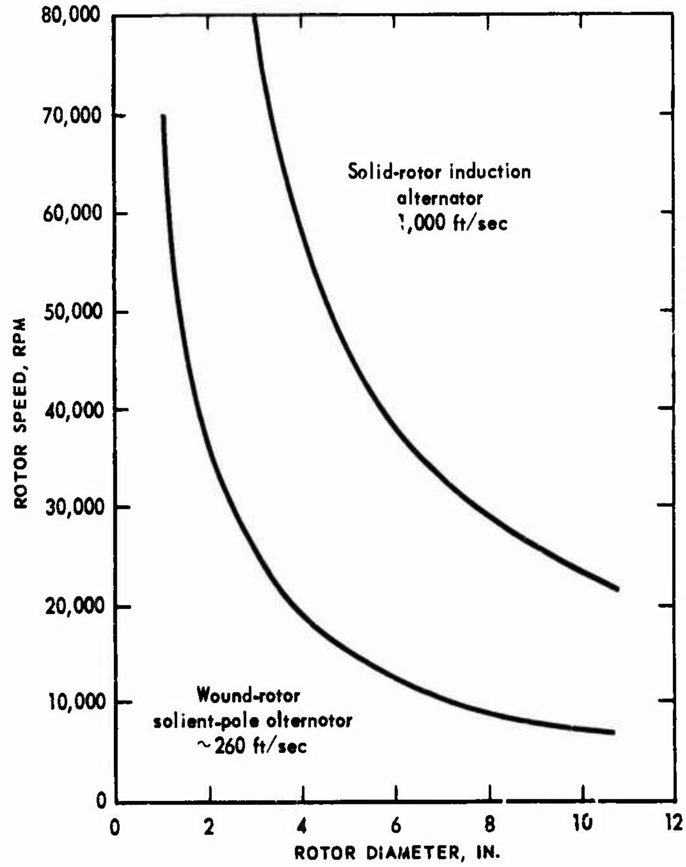


Fig. II-57—Rotor Speed as a Function of Rotor Diameter

Advantages. The advantages of an electric-drive system are:

(a) The designer is provided with greater flexibility for locating vehicle components.

(b) Vehicle speed may be operator-varied independent of engine speed.

(c) Faster vehicle response and greater agility are provided.

(d) A smooth variable-controlled vehicle speed is obtained.

(e) Vehicle dynamic braking and regenerative steering are provided.

(f) Simpler and easier operator controls result.

(g) A variable speed-control ratio for tracked vehicles is provided.

(h) Ready power to all wheels is provided.

(i) Vehicle mobility is increased.

(j) Engine operation is possible at the most economical power range.

(k) The capability to provide auxiliary power or prime electrical power as a mobile generating station is offered.

Disadvantages. The disadvantages of an electric-drive system are:

(a) Efficiencies lower than those of comparable mechanical or hydrokinetic systems.

(b) Greater size and weight than that of comparable mechanical or hydrokinetic systems.

(c) Higher cost than that of present mechanical or hydrokinetic systems.

(d) Unproved reliability in tactical vehicles (may achieve the reliability of present systems).

PREDICTIONS

It is predicted that technological advancements will be made in both ac and dc electric-drive systems (for use in tactical vehicles) if R&D by the US Army is continued. These advancements will improve the capabilities of special types of tactical vehicles in which difficulty in meeting the physical performance characteristics (when more conventional drive systems were used) has been encountered.

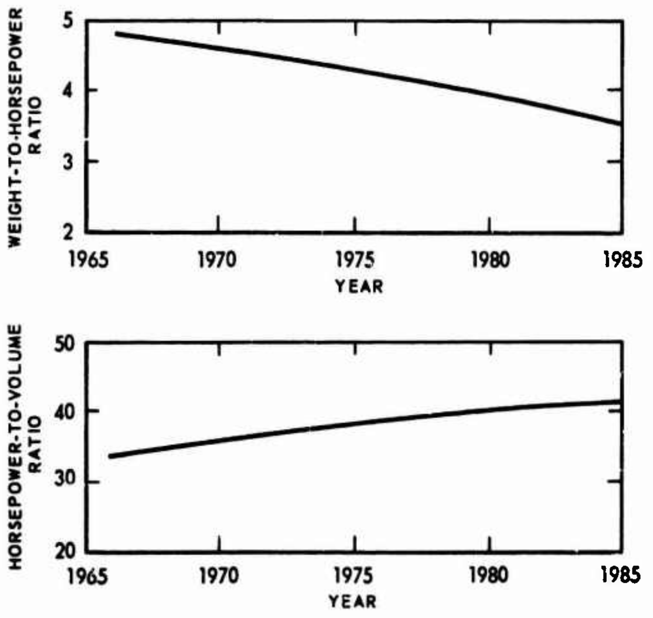


Fig. II-58—Predicted Specific Ratio Improvements of Ac Electric-Drive Systems for 50-ton Tracked Vehicle

Final drives and electric cabling not included.

Specific weight and size (volume) to horsepower ratios will improve for dc electric-drive systems, and an even more significant improvement will be noted in the ac electric-drive systems, as illustrated in Fig. II-58. These improvements could be realized through future development of higher-speed motors and/or generators, miniaturized components, lighter-weight materials, and an electronic shifting device to engage the gears of a mechanical 2-speed reducer and increase the torque range of the ac and dc electric-drive systems at low speed.

Development affecting the improvement of the ac electric-drive systems only will include the simplification of ac system control circuitry, motors and alternators that can operate at higher frequencies, and an alternator for an ac electric-drive system that can be incorporated in a gas-turbine engine.

Efficiency

The efficiency of both the dc and ac electric-drive systems will improve as illustrated in Fig. II-59. The improvements realized through future electric-drive-system development will include a reduction in losses owing to windage of rotating equipment for both dc and ac systems (more significantly for the ac electric-drive system); and improved cooling techniques for both types of system. The ac system will experience improved power factors, improved alternators, and better electrical steels that will increase flux densities and reduce hysteresis losses.

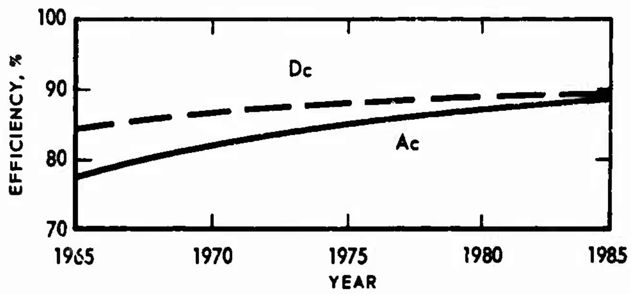


Fig. II-59—Predicted Improvement in the Efficiency of Ac and Dc Electric-Drive Systems

Reliability and Maintainability

The reliability and maintainability of both the dc and ac electric-drive systems will improve through better methods of system waterproofing. Also fungus-resisting components will be developed for ac, and more significantly, for dc electric-drive systems. The ac system will benefit from the incorporation of fewer parts. Further, silicon-controlled rectifiers that will operate at higher temperatures, carry increased current, and operate at higher voltages will be used in the ac system.

The reduction of weight and size (volume) of electric-drive systems will enable developers to produce smaller and lighter vehicles in the higher power ranges than those now produced with mechanical-drive systems. The reduction of vehicle weight will reduce the ground pressures exerted by the vehicle track or wheels and will increase vehicle mobility. Improved electric-drive-system efficiency and reliability and maintainability, as outlined above, will enable economical vehicle operation and extend vehicle operating range.

Dc electric-drive systems developed by industry for commercial vehicles are not required to operate under environmental conditions as severe as are tactical-vehicle electric-drive systems. Numerous electric systems are available at this time for commercial-vehicle applications, for vehicles not required to undergo severe environmental operating conditions. Dc electric-drive systems now available in the commercial market should be modified to be acceptable for small battery-powered limited-range tactical vehicles for which silent operation is required and with which freedom from noxious fumes is desirable. In addition, dc electric-drive systems now are used with success in large vehicles powered by reciprocating engines. In this application each wheel of the vehicle is individually powered, and excellent tractive effort, dynamic braking, and ease of control are provided. These dc electric-drive systems should be modified for use in special tactical vehicles that are powered by reciprocating engines and where mechanical drives cannot be readily installed to furnish power to the wheels or track sprockets. This development could be accomplished in the near future.

Considerable developmental effort has been expended to provide sufficient data to develop an acceptable ac electric-drive system for tactical vehicles. Additional data are required, and it is not foreseen that this effort will be conducted by industry since industry's requirement for such a system is limited. Acceptable ac electric-drive systems have not been developed by industry for either commercial-or tactical-vehicle applications. Ac electric-drive systems have a greater potential than comparable dc systems but ac system development cannot be accomplished before the 1975–1985 time frame. A vehicle should be designed specifically for an ac electric-drive system to take full advantage of this kind of system. This technology and the data obtained from use then could be applied to other special tactical vehicles.

The ac electric-drive system appears to have the potential for successful development for use in special-purpose or amphibious tactical vehicles, vehicles in which wheels must be individually powered and in which mechanical drives are difficult to incorporate. Ac drives also show promise for application in articulated vehicles where the wheels or tracks of more than one section must be powered. In addition, ac drives could be applied to large tracked vehicles that have large power requirements and, should a gas turbine be incorporated, the drive could readily be coupled to the engine. This type of system would lend itself to vehicles requiring a low silhouette. Successful development of an ac electric-drive system, if initiated in the near future, could be accomplished within the 1975–1985 time frame.

CONCLUSIONS

US Army R&D of electric-drive systems <u>is</u> warranted. Both the ac and dc electric-drive systems could improve the physical and performance characteristics of some types of tactical vehicles.

Dc electric-drive systems are being developed by industry for commercial vehicles that are not required to operate under as severe environmental conditions as are tactical vehicles. Available commercial dc electric-drive systems should be modified to be acceptable for small battery-powered limited-range tactical vehicles for which silent operation is a requirement. Dc systems also should be modified for special tactical vehicles powered by reciprocating engines and in which mechanical drives cannot be readily installed to provide power to the wheels or track sprockets. This development could be accomplished in the 1966–1975 time frame.

Acceptable ac electric-drive systems have not been developed by industry for either commercial-or tactical-vehicle applications. Ac electric-drive systems have the potential of reducing the weight and size of a tactical vehicle over comparable dc electric-drive systems, but the development could not be accomplished before the 1975–1985 time frame. A vehicle should be specifically designed to take full advantage of an ac electric-drive system, and this technology then could be applied to other special tactical vehicles.

PART III

Applicability, Compatibility, and Potential Contributions To Tactical Vehicles

Chapter 22

DISCUSSION

Many of the energy- and power-conversion devices evaluated during this study that were already developed or are currently being developed are not applicable for use in future tactical vehicles. On the other hand, many other devices that were developed in the past and those currently in research or under development will be applicable for use in future tactical vehicles.

Despite technological advances in this field, some of the devices for one or many reasons will not be applicable for use in future tactical vehicles since they cannot surpass the capability of devices currently available.

Energy- and power-conversion devices that are not applicable for use in tactical vehicles within the foreseeable future are:

Energy-Conversion Devices Not Applicable

(a) Steam engine, reciprocating
(b) Steam engine, turbine
(c) Spark-ignition engine, ammonia fueled
(d) Compression-ignition engine, ammonia fueled
(e) Fuel cell
(f) Battery, primary or secondary
(g) Battery, electrochemical energy-storage system
(h) Nuclear reactor
(i) Unique energy-conversion devices
(j) KGG cycle (Kuhns) engine

Power-Conversion Devices Not Applicable

(a) Progressive sliding gear
(b) Selective sliding gear
(c) Constant mesh
(d) Friction drive
(e) "Hydramatic"
(f) "Torqmatic"

Energy- and power-conversion devices that are applicable and are now available for use in tactical vehicles or will be within the foreseeable future are:

Applicable Energy-Conversion Devices

(a) Conventional spark-ignition engines
(b) Conventional compression-ignition engines

Applicable Power-Conversion Devices

Device	Wheeled	Tracked
(a) Synchromesh	x	x
(b) Torque converter	x	
(c) Torque converter, planetary gear, TX series	x	x
(d) Torque converter, planetary gear, CD series		x
(e) Torque converter, planetary gear, XT series		x
(f) Torque converter, planetary gear, XTG series		x
(g) Torque converter, planetary gear, X series		x

Energy- and power-conversion devices that would be applicable but require R&D for tactical vehicles fielded in the 1966–1975 time frame are:

Energy-Conversion Devices Requiring R&D, for 1966–1975

(a) Rotary spark-ignition engine
(b) Dynastar spark-ignition engine
(c) Dynastar compression-ignition engine
(d) Rotary hybrid engine
(e) Hybrid engine
(f) Stirling-cycle "Dineen" process engine
(g) Differentially supercharged engine
(h) VCR turbine engine
(i) VHO turbine engine
(j) Piston-turbine compound engine

Power-Conversion Devices Requiring R&D

(a)* Torque converter, planetary gear, TX series (wheeled)
(b)* Torque converter, planetary gear, X series (tracked)
(c) Belt drive
(d) Hydrostatic (narrow speed-torque range) drive
(e) Electric (dc) drive

Energy- and power-conversion devices that would be applicable but require R&D for tactical vehicles fielded in the 1975–1985 time frame are:

Energy-Conversion Devices Requiring R&D, for 1975–1985

(a) EHO engine
(b) Free-piston turbine engine
(c) Gas turbine, single shaft
(d) Gas turbine, two shaft
(e) Gas turbine, differential

Power-Conversion Devices

(a) Hydromechanical transmission
(b) Electric (ac) drive
(c) Hydrostatic (medium speed—torque range) drive

*Available now but does not cover entire power range for tactical vehicles and has potential of further improvements.

After delineating nonapplicable energy- and power-conversion devices, all those remaining were evaluated for their compatibility and suitability for various types of vehicle within their respective horsepower ranges (see Table III-1).

Many of the evaluated energy- and power-conversion devices could be developed enough to improve the capability of tactical vehicles by replacing their energy- and power-conversion devices. However, Government support for their development is required since industry has little or no application for these devices in commercial vehicles. Therefore industry has no incentive to develop these devices for application to tactical vehicles with their own funds. The new devices, if successfully developed, would contribute to the Army's effort of achieving improved tactical vehicles in the future. These devices and their potential contributions to tactical vehicles are presented in Table III-2.

TABLE III-1

Compatible Energy- and Power-Conversion Devices with Applicable Type of Vehicle and Horsepower Range

Energy-conversion device	Power-conversion device	Wheeled, hp range	Tracked, hp range	Amphibious, hp range		Special purpose, hp range	
				Wheeled	Tracked	Wheeled	Tracked
Conventional spark-ignition engine	Synchromesh transmission	0–120	0–120	0–120	0–120	0–120	0–120
	Belt drive	0–120	NA[a]	NA	NA	NA	NA
	Torque converter	0–250	NA	NA	NA	NA	NA
	TCPG[b] TX series	0–250	NA	0–250	0–250	0–250	NA
	TCPG X series	NA	0–250	NA	0–250	NA	0–250
	Hydromechanical transmission	0–250	0–250	0–250	0–250	0–250	0–250
	Hydrostatic drive	0–120	0–120	0–250	0–120	0–250	0–250
	Electric drive	0–120	0–120	0–250	0–120	0–250	0–250
Rotary spark-ignition engine	Synchromesh transmission	0–120	0–120	0–120	0–120	0–120	0–120
	Belt drive	0–120	NA	NA	NA	NA	NA
	Torque converter	0–250	NA	NA	NA	NA	NA
	TCPG TX series	0–250	NA	0–250	0–250	0–250	NA
	TCPG X series	NA	0–250	NA	0–250	NA	0–250
	Hydromechanical transmission	0–250	0–250	0–250	0–250	0–250	0–250
	Hydrostatic drive	0–120	0–120	0–250	0–120	0–250	0–250
	Electric drive	0–120	0–120	0–250	0–120	0–250	0–250
Dynastar spark-ignition engine	Synchromesh transmission	0–120	0–120	0–120	0–120	0–120	0–120
	Belt drive	0–120	NA	NA	NA	NA	NA
	Torque converter	0–250	NA	NA	NA	NA	NA
	TCPG TX series	0–250	NA	0–250	0–250	0–250	NA
	TCPG X series	NA	0–250	NA	0–250	NA	0–250
	Hydromechanical transmission	0–250	0–250	0–250	0–250	0–250	0–250
	Hydrostatic drive	0–120	0–120	0–250	0–120	0–250	0–250
	Electric drive	0–120	0–120	0–250	0–120	0–250	0–250
Conventional compression-ignition engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	120–1000	NA	120–1000	120–1000	120–1000	NA
	TCPG X series	NA	120–1000	NA	120–1000	NA	120–1000
	Hydromechanical transmission	120–1000	120–1000	120–1000	120–1000	120–1000	120–1000
	Hydrostatic drive	NA	NA	120–1000	NA	120–1000	120–1000
	Electric drive	500–1000	500–1000	120–1000	500–1000	120–1000	120–1000

TABLE III-1 (continued)

Energy-conversion device	Power-conversion device	Wheeled, hp range	Tracked, hp range	Amphibious, hp range		Special purpose, hp range	
				Wheeled	Tracked	Wheeled	Tracked
Dynastar compression-ignition engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	120–1000	NA	120–1000	120–1000	120–1000	NA
	TCPG X series	NA	120–1000	NA	120–1000	NA	120–1000
	Hydromechanical transmission	120–1000	120–1000	120–1000	120–1000	120–1000	120–1000
	Hydrostatic drive	NA	NA	120–1000	NA	120–1000	120–1000
	Electric drive	500–1000	500–1000	120–1000	500–1000	120–1000	120–1000
Rotary hybrid engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	120–500	NA	120–500	120–500	120–500	NA
	TCPG X series	NA	120–500	NA	120–500	NA	120–500
	Hydromechanical transmission	120–500	120–500	120–500	120–500	120–500	120–500
	Hydrostatic drive	NA	NA	120–500	NA	120–500	120–500
	Electric drive	NA	NA	120–500	NA	120–500	120–500
Hybrid engine	Synchromesh transmission	0–120	NA	0–120	NA	0–120	NA
	Belt drive	0–120	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	0–1000	NA	0–1000	0–1000	0–1000	NA
	TCPG X series	NA	0–1000	NA	0–1000	NA	0–1000
	Hydromechanical transmission	0–1000	0–1000	0–1000	0–1000	0–1000	0–1000
	Hydrostatic drive	0–120	0–120	0–1000	0–120	0–1000	0–1000
	Electric drive	0–120; 500–1000	0–120; 500–1000	0–1000	0–120; 500–1000	0–1000	0–1000
Stirling-cycle Dinnen process engine	Synchromesh transmission	0–120	NA	NA	NA	NA	NA
	Belt drives	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	0–120	NA	NA	NA	NA	NA
	TCPG X series	NA	NA	NA	NA	NA	NA
	Hydromechanical transmission	0–120	NA	NA	NA	NA	NA
	Hydrostatic drive	0–120	NA	NA	NA	NA	NA
	Electric drive	0–120	NA	NA	NA	NA	NA
Differentially supercharged engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	120–1000	NA	120–1000	120–1000	120–1000	NA
	TCPG X series	NA	120–1000	NA	120–1000	NA	120–1000
	Hydromechanical transmission	120–1000	120–1000	120–1000	120–1000	120–1000	120–1000
	Hydrostatic drive	NA	NA	120–1000	NA	120–1000	120–1000
	Electric drive	500–1000	500–1000	120–1000	500–1000	120–1000	120–1000
VCR engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	250–1000	NA	250–1000+	250–1000+	250–1000+	NA
	TCPG X series	NA	250–1000+	NA	250–1000+	NA	250–1000+
	Hydromechanical transmission	250–1000	250–1000+	250–1000+	250–1000+	250–1000+	250–1000+
	Hydrostatic drive	NA	NA	250–1000+	NA	250–1000+	250–1000+
	Electric drive	500–1000	500–1000+	250–1000+	500–1000+	250–1000+	250–1000+
VHO engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	250–1000	NA	250–1000+	250–1000+	250–1000+	NA
	TCPG X series	NA	250–1000+	NA	250–1000+	NA	250–1000+

TABLE III-1 (continued)

Energy-conversion device	Power-conversion device	Wheeled, hp range	Tracked, hp range	Amphibious, hp range		Special purpose, hp range	
				Wheeled	Tracked	Wheeled	Tracked
	Hydromechanical transmission	250–1000	250–1000+	250–1000+	250–1000+	250–1000+	250–1000+
	Hydrostatic drive	NA	NA	250–1000+	NA	250–1000+	250–1000+
	Electric drive	500–1000	500–1000+	250–1000+	500–1000+	250–1000+	250–1000+
EHO engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	500–1000	NA	500–1000+	500–1000+	500–1000+	NA
	TCPG X series	NA	500–1000+	NA	500–1000+	NA	500–1000+
	Hydromechanical transmission	500–1000	500–1000+	500–1000	500–1000+	500–1000+	500–1000+
	Hydrostatic drive	NA	NA	NA	NA	NA	500–1000+
	Electric drive	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
Free-piston turbine engine	Synchromesh transmission	120–1000	120–1000	120–1000	120–1000	120–1000	120–1000
	Belt drive	NA[a]	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG[b] TX series	120–1000	NA	120–1000	120–1000	120–1000	NA
	TCPG X series	NA	120–1000	NA	120–1000	NA	120–1000
	Hydromechanical transmission	120–1000	120–1000	120–1000	120–1000	120–1000	120–1000
	Hydrostatic drive	120–1000	NA	NA	NA	NA	120–1000
	Electric drive	120–1000	500–1000	120–1000	500–1000	120–1000	120–1000
Piston-turbine compound engine	Synchromesh transmission	NA	NA	NA	NA	NA	NA
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	120–1000	NA	120–1000	120–1000	120–1000	NA
	TCPG X series	NA	120–1000	NA	120–1000	NA	120–1000
	Hydromechanical transmission	120–1000	120–1000	120–1000	120–1000	120–1000	120–1000
	Hydrostatic drive	NA	NA	120–1000	NA	120–1000	120–1000
	Electric drive	500–1000	500–1000	120–1000	500–1000	120–1000	120–1000
Gas-turbine single-shaft engine	Synchromesh transmission	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	NA	NA	NA	NA	NA	NA
	TCPG X series	NA	NA	NA	NA	NA	NA
	Hydromechanical transmission	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
	Hydrostatic drive	NA	NA	500–1000+	NA	500–1000+	500–1000+
	Electric drive	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
Gas-turbine two-shaft engine	Synchromesh transmission	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	NA	NA	NA	NA	NA	NA
	TCPG X series	NA	NA	NA	NA	NA	NA
	Hydromechanical transmission	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
	Hydrostatic drive	NA	NA	500–1000+	NA	500–1000+	500–1000+
	Electric drive	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
Gas-turbine differential engine	Synchromesh transmission	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
	Belt drive	NA	NA	NA	NA	NA	NA
	Torque converter	NA	NA	NA	NA	NA	NA
	TCPG TX series	NA	NA	NA	NA	NA	NA
	TCPG X series	NA	NA	NA	NA	NA	NA
	Hydromechanical transmission	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+
	Hydrostatic drive	NA	NA	500–1000+	NA	500–1000+	500–1000+
	Electric drive	500–1000	500–1000+	500–1000+	500–1000+	500–1000+	500–1000+

[a]NA, not applicable.
[b]TCPG, torque converter, planetary gear.

TABLE III-2

Energy- and Power-Conversion Devices and Their Potential Contributions to Tactical Vehicles

Device	Contribution to physical improvement of vehicle	Contribution to operational improvement of vehicle
	Energy-Conversion Devices	
Rotary spark-ignition engine	Reduces vehicle weight, offers more space for cargo or personnel	Decreases operational cost; decreases vehicle maintenance requirements; reduces logistic requirements
Dynastar compression ignition engine	Reduces vehicle weight; offers more space for cargo or personnel	Decreases operational cost; complies with fuel policy
VCR engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
VHO engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
EHO engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Hybrid engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy; contributes prompt mobilization capacity
Stirling-cycle engine	—	Offers silent operation
Gas-turbine engine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
Piston-turbine compound engine	Reduces vehicle weight; offers more space for cargo and personnel	Improves mobility; decreases operational cost; complies with fuel policy
Free-piston turbine	Reduces vehicle weight; offers more space for cargo and personnel	Decreases operational cost; complies with fuel policy
	Power-Conversion Devices	
Torque converter, planetary gear, TX series	Reduces vehicle weight	Improves mobility; improves controllability; decreases logistic requirements; decreases vehicle maintenance requirements; increases vehicle reliability; decreases operational cost
Torque converter, planetary gear, X series	Reduces vehicle weight; offers more space for cargo and personnel	Improves mobility; improves controllability; decreases logistic requirements; decreases vehicle maintenance requirements; increases vehicle reliability; decreases operational cost
Belt drive	Provides better load distribution on axles; increases vehicle ground clearance; readily adaptable to various small engines	Decreases logistic requirements
Hydrostatic transmission	Offers flexibility of design; provides better load distribution on axles and increased ground clearance for wheeled vehicles; provides more space for cargo and personnel; readily adaptable to various engines	Improves mobility and ease of operation; improves controllability; reduces vehicle maintenance requirements; decreases logistic requirements; reduces operational cost

TABLE III-2 (continued)

Device	Contribution to physical improvement of vehicle	Contribution to operational improvement of vehicle
Hydromechanical transmission	Reduces vehicle weight; offers more space for cargo and personnel; readily adaptable to various engines	Improves mobility and ease of operation; improves controllability; reduces vehicle maintenance requirements; decreases logistic requirements; reduces operational cost
Electric drive	Offers flexibility of design; provides better axle loading ratios and increased ground clearance; provides more space for cargo and personnel; readily adaptable to various engines	Improves mobility and ease of operation; provides better vehicle control; reduces vehicle maintenance; decreases logistic requirements; reduces operational cost

PART IV

Tradeoff Analysis and Recommended Programs

Chapter 23

INTRODUCTION

There are many technically feasible energy- and power-conversion devices that could, if developed, improve the capabilities of future tactical vehicles. Technical achievement alone, however, may not justify a recommended expenditure of military R&D funds. To warrant development a device often must satisfy numerous criteria and always satisfy at least one critical demand. A careful analysis of the factors that determine the overall contribution of any given device is required before the decision to recommend a device is reached. Such decisions are reached not only through extensive evaluation of the advantages and disadvantages offered by the device itself but through a determination of the ultimate advantages accruing from device development and usage. To achieve this goal a tradeoff analysis was conducted. A systematic elimination of those devices not meeting minimal requirements of a comprehensive checklist of "influencing factors" represents a most important aspect of the selection-for-recommendation process.

The factors considered to wield a major influence in the tradeoff analysis performed for this study are:

(a) Technical contribution of device
(b) Types of vehicles for which the devices are applicable
(c) Future density of tactical vehicles for which the devices have applicability
(d) Probability of acceptable development
(e) Time span required to achieve success
(f) Cost of development programs
(g) Unit cost of device in production quantities
(h) Ability of industry to produce devices at time of full mobilization

A survey of all known energy- and power-conversion devices was made, and the findings have been presented and discussed in the earlier phases of this report. Only those devices showing promise to advance the state of the art are evaluated in Part IV.

Spark-ignition reciprocating gasoline engines were predominantly used for the propulsion of military vehicles before and during WWII when industry was required to produce large quantities of reliable vehicles in a relatively short period. At that time, gasoline engines and the mechanical transmissions were in common use for commercial passenger cars, trucks, and off-highway vehicles. Most development efforts were directed toward making these commercial units suitable for use in military vehicles. The capacity of industry to mobilize for production lay in spark-ignition reciprocating gasoline engines and mechanical transmissions or power trains. The tooling, special equipment, and personnel experienced to react quickly to the needs of the military were available. Further the fuel policy of the military effectively prohibited the use of diesel engines. The use of turbine-engine-type fuels for military vehicles was not sanctioned.

In 1961 a change was made in the fuel policy of the military. The use of diesel engines of 200 hp and over was sanctioned for military vehicle application. Diesel engines below 200 hp could not compete on the basis of cost, both initial and operating, with the automotive-type gasoline engines in production. This is still true at time of writing and will remain so in the foreseeable future.

So-called "automatic" transmissions have been under development by industry for commercial applications, and these torque-converter planetary-gear transmissions or power trains have also been developed for military vehicles. These developments, applicable for more than one power range and vehicle configuration, encompassed a long time period. New tactical vehicles envisioned by the US Army require greater acceleration and agility, and thus a requirement exists for greater engine output without increasing the weight and size of past engines.

Diesel engines are receiving greater acceptance for commercial vehicle applications, and, as production quantities increase, the initial unit cost decreased. Diesel engines have undergone a continuing improvement in performance, engine life, and weight-(or size)-to-horsepower ratio. Although the initial cost of diesel engines will remain higher than that of spark-ignition engines, the overall cost may be lower when engine life, reliability, maintainability, and operating costs are considered.

Fuel efficiency is a very important facet of military vehicle operation. The range of a military vehicle and the logistic support required for such vehicle are adversely affected by inefficient fuel consumption. Conversely, vehicle range may be extended, and the logistic support considerably reduced through efficient fuel consumption. Although the cost of fuel at a military depot may be 13 cents a gallon, this cost may increase to reach $2 or $3 per gallon before the fuel reaches a using vehicle. For these reasons, diesel engines over 200 hp now are used in some military tactical vehicles. However, the diesel engines of today are physically too large and heavy for use in many tactical vehicles. New lightweight and smaller engines are being developed to overcome these deficiencies. Compatible transmissions and power trains also are being developed to meet tactical-vehicle operational requirements and to reduce device weight and size.

The present state of the art of gas-turbine engines for vehicular usage approaches that of diesel engines in the late nineteen thirties. Although very successful in aircraft and stationary equipment applications, the gas-turbine engines are far from acceptable for use in tactical vehicles. Industry has made some attempts to develop the gas-turbine engine for use in commercial vehicles. The high cost for this development is recognized as is the fact that the initial market is limited to trucking concerns. The first commercial application will be in transcontinental trucks that require engines having at least 500 hp and that operate 80 to 90 percent of the time at maximum permissible highway speeds. The first military application of the gas-turbine engines will be in heavy tracked vehicles where engine weight and size are limited. The Army's new main battle tank requires a 1500-hp engine with a compatible power train that can operate within the confines of a space previously occupied by an engine of only 1000 hp and a related power train. The gas-turbine engine has a strong potential for meeting the space requirement of the new main battle tank, but the cost of the engine in production quantities will be considerably higher than that of a comparable diesel engine.

Most companies recognize the elements of risk inherent in the development of a gas-turbine engine for commercial applications. Should success in the engine development be achieved it is possible that the costs of such an engine would be found to be higher than the commercial sales market would permit. Cost may not be a prime factor in the procurement of such engines for military applications as there may not be an alternative device available to meet the physical and performance characteristics of the vehicle for which the engines are being procured.

The most promising gas-turbine-engine development sponsored by the military appears to be the 1500-hp engine designed for use in a main battle tank. The Army has specified that this new regenerative gas-turbine engine must be designed for production at a unit cost of $15 per horsepower at a nominal rate of 1000 units per year. In a prior 600-hp competitive gas-turbine development program, a design goal of $20 per horsepower was specified.

The diesel AVDS 1790-2 engine used in the M60 tank was purchased at a unit price of approximately $17,000. The production rate was 720 units per year. The net power output of this engine is slightly over 600 hp. Should development of the 1500-hp gas-turbine engine be successful and a life expectancy equal to or greater than the number of hours of life of a diesel engine be achieved, high development costs can be justified. The overall operating cost of a 1500-hp gas-turbine engine would compare favorably with the costs of a diesel engine rendering a like power output.

A list of the tactical vehicles envisioned by the Army for future use was prepared to determine the impact such vehicles would have on Army requirements for energy- and power-conversion devices. This list was used to determine anticipated tactical-vehicle densities by vehicle type, weight, and power class, and by applicable type of energy- or power-conversion device. However, the list was not included in the report since this information would have required security classification. Consideration was given to the potential number of specific devices required in determining development time, unit production cost, maintenance cost, and logistics. Anticipated relative densities of wheeled and tracked vehicles, by power class, are shown in Fig. IV-1.

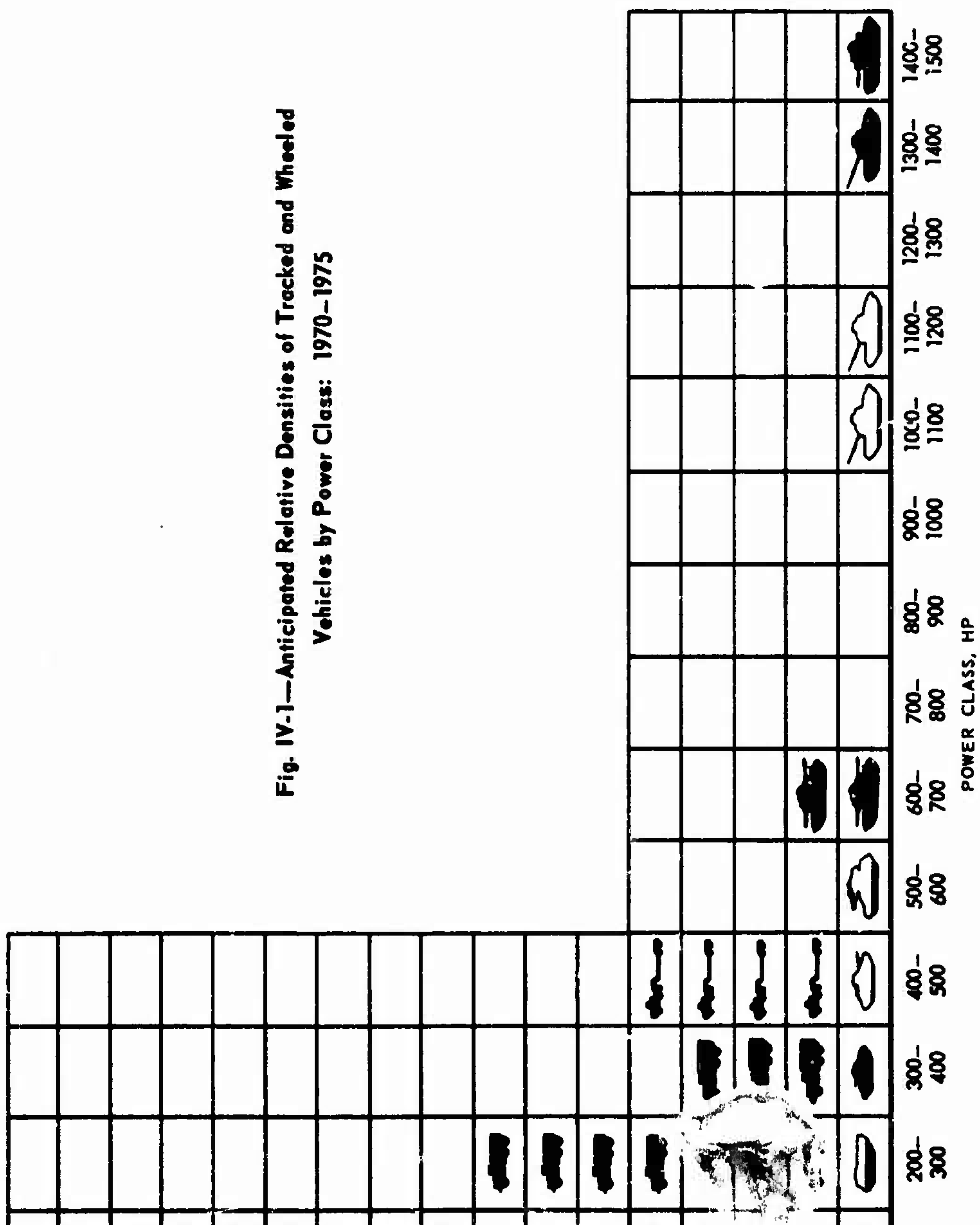


Fig. IV-1—Anticipated Relative Densities of Tracked and Wheeled Vehicles by Power Class: 1970–1975

Chapter 24

COST ANALYSIS AND EVALUATION

INTRODUCTION

Cost is a major consideration when determining what energy- and power-conversion devices should be developed by the Army. Both device development and the unit cost in production quantities must be considered. Members of the study group contacted and visited persons at industrial plants, as well as at Government agencies, to obtain the historical cost data of similar programs. In most instances there was a considerable reluctance on the part of the persons visited or contacted to discuss specific cost data. Individuals in industry, in particular, were highly sensitive regarding the disclosure of any cost figures because such information is considered to be of a proprietary nature. Personnel at Government agencies stated that most cost data available from the agencies were not very meaningful, in that costs other than those defined in the original work scope of a program were included. Members of the study group were compelled to draw on personal knowledge and experience gained while working in industry, literature search, and an evaluation of that scant amount of information available from the Government and industrial sources.

MARKET ANALYSIS AND COST ESTIMATES

Many variables that affect costs must be considered before a reasonable cost estimate may be made. The accepted procedure in industry before making a cost estimate is to conduct a somewhat extensive market analysis. The market analysis is directed to a detailed consideration of primary who-, what-, and why-type questions, i.e., Who is the customer? What are the requirements, incentives, and competition? What facilities and personnel are required? Complete answers to the eight groups of survey questions listed below furnish the data base necessary for comprehensive analysis.

(a) Who is the customer? What are his procurement requirements?
  (1) What will the quantity of the initial purchase be?
  (2) What is the potential for follow-on orders?
  (3) What quantities of spares will be required?
  (4) Will the contract be modified to include other requirements?

(b) What incentives to bid for a contract exist?
(1) What is the potential commercial market?
(2) Will the company retain proprietary rights to the item produced?
(3) Will possible by-products produce patentable items?
(4) What is the margin of profit?
(c) What is the competition?
(1) How many, and who, are the competitors?
(2) What advantages do the competitors have?
(3) Does the competitor know his costs?
(4) Is the competitor willing to risk loss?
(d) What is the attitude of management and the corporate philosophy?
(1) How will the contract affect the corporate image?
(2) Is management willing to incur risk?
(3) Does management intend to continue to expend effort in this area?
(e) What are the tool and facility requirements?
(1) Are present tools and facilities appropriate?
(2) Will present tools and facilities be available?
(3) Can new tools and facilities acquired for the contract be used for other company purposes or products?
(f) What are the personnel requirements?
(1) What personnel will be available to perform the work required by the contract?
(2) Can additional qualified personnel be obtained?
(3) Will the contract enable personnel to keep abreast of developments in areas of work important to both personnel and the company?
(g) What does the customer demand?
(1) Does the customer hold realistic expectations?
(2) Does the customer understand the impact of his specifications?
(3) Does the customer exhibit a willingness to negotiate to circumvent insurmountable problems?
(h) What are the basic direct costs?
(1) What are the costs of the materials required?
(2) What is the cost of the direct labor?
(3) What is the cost of support labor?
(4) Do the terms of the contract permit the prorating of prior research costs and if so, how may this best be accomplished?

Administrative Costs

In addition to industry's cost for development or production programs for energy- or power-conversion devices, there are administrative costs incurred by the cognizant Government agency. The Government costs vary not only with the size of the contract but also are affected proportionally by the duration of time required by the contractor to consummate a program. Although Government administrative costs are not included in the estimated cost figures, these costs, for the purpose of performing a tradeoff analysis, are considered proportional to the cost of any development program.

The development of a device may continue for application to more than one class or horsepower range of vehicle. In this case the succeeding development costs will be reduced somewhat since many of the technological advancements will be applicable to all ranges. A typical cost estimate for production of a 300-hp internal-combustion compression-ignition engine in various quantities is shown in Table IV-1. Data presented in this table are representative of a large reservoir of data available to designers of compression-ignition engines for tactical vehicles. Units having 600 hp have been developed and are used with success in commercial trucks. Units to 800 hp have been developed and used with success in military vehicles. However, as horsepower requirements increase, there is further departure from the present state of the art,and available data become less applicable to the development of larger engines.

TABLE IV-1

Production Cost Estimate for 300-hp Internal-Combustion Compression-Ignition Engine

Item	Total cost, dollars	Production run, units		
		1000	2000	10,000
		Cost per unit, dollars		
Production design, engineering	700,000	700	350	70
Materials	1,200	1200	1100	1000
Direct shop labor	680[a]	680	600	580
Overhead at 200%	—	1360	1200	1160
Tooling	1,600,000	1600	800	160
Special equipment	400,000	400	200	60
Manuals	90,000	90	45	10
Subtotal	—	6030	4295	3040
General administration at 10%	—	603	430	304
Subtotal	—	6633	4725	3344
Profit at 7%	—	465	330	234
Total cost per unit	—	7098	5055	3578

[a]From 80 hr at $8.50 per hr.

To date there are no acceptable gas-turbine engines for either commercial or military vehicles. Consequently few historical cost data are available. Data available on the gas-turbine engines used for aircraft and stationary equipment has little applicability to tactical vehicles because the conditions of operation present drastic differences. After development of the first gas-turbine engine for a tactical vehicle in any horsepower range, data will be available for design and cost estimates for the development of the gas-turbine engine in most horsepower ranges. There is a progressive reduction in the cost of each successive development.

Estimated Development Costs

The estimated development costs for both the gas-turbine and the compression-ignition reciprocating engine as a function of respective horsepower sizes were prepared and are shown in Fig. IV-2. The unit costs (assuming

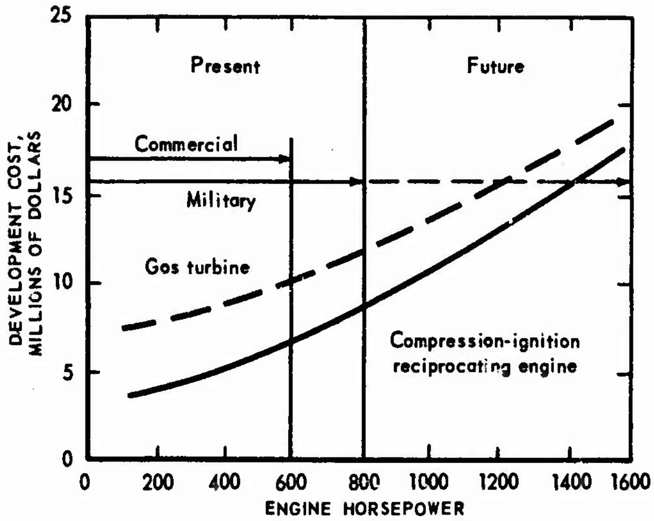


Fig. IV-2—Development Cost of Engines as a Function of Horsepower Output

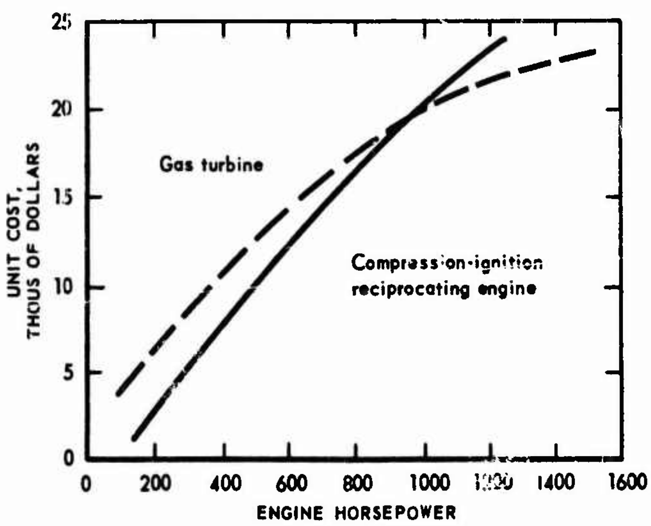


Fig. IV-3—Unit Requisition Cost of Engines as a Function of Horsepower Output production quantities) for both gas-turbine engines and the compression-ignition reciprocating engines as a function of horsepower size are shown in Fig. IV-3.

Estimated costs of developing or producing potential energy- or power-conversion devices have been based, to a great extent, on the relative costs incurred in the development and production of similar devices. Lead time to successful device development depends in large measure on the technological status of promising devices, the approaches planned for overcoming deficiencies, and the interest of industry in the final product.

The hybrid, Dynastar, rotary-piston, and Stirling-cycle engines, in principle, have a very close relation with present-day reciprocating engines. The compound and free-piston engines have features relating to gas-turbine engines. Transmissions and power trains of the hydrokinetic group have been developed with success, and the cost data acquired through this experience can be related to other power-conversion devices showing potential for development. The hydromechanical transmissions, or power trains, although differing in principle from the hydrokinetic devices, appear to require a like developmental effort. Much more difficult to estimate are the developmental costs of both hydrostatic and electric-drive devices, since the ultimate device developed depends on the requirements imposed by the vehicle in which the device will be used. In each case of energy- and power-conversion-device development, those devices offering the greatest potential for technological advancement and vehicle improvement through development and use have the least prior and related cost data experience available as a guideline.

Horsepower class and probable vehicle density are the basis for the order of listing of the device-cost estimates prepared for this report. Every effort was made to establish accurate cost estimates; however, these cost figures were used only to ascertain the relative impact of these costs in the tradeoff analysis.

Development Time and Risk

There is an element of risk in most development programs of achieving success within the specified time and budget. The most successful development programs are initiated on a plan of sound principles and concepts, but the plan of performance of any program must be revised as elements of the program are achieved or as deficiencies are encountered. Infallible assurance that a planned program can be accomplished as estimated cannot be given.

Tables IV-2 and IV-3 present a detailed estimate of development time required for the successful development of typical energy- and power-conversion devices. An estimate of the probability of success of various energy- and power-conversion devices within a given number of years is presented in Table IV-4.

Mobilization Capacity of Industry

An effort was made to determine industry's capacity to produce various energy- and power-conversion devices when full mobilization is required, as in time of war. Industry could produce all required spark-ignition reciprocating engines and mechanical transmissions in a relatively short time by the simple expedient of diverting commercial production to military production.

TABLE IV-2

Development Time Required for Energy-Conversion Devices[a]

(In years)

Phase	Reciprocating spark-ignition engine	Reciprocating compression-ignition engine	Reciprocating hybrid engine	Dynostar engine	Rotary-piston engine	Stirling-cycle engine	Gas-turbine engine	Compound engine	Free-piston-turbine engine
Research	0	0	2.0	0	1.0	1.0	1.0	1.0	2.0
Development design	0	1.0	1.0	0.9	1.0	2.0	2.0	1.0	1.6
Prototype construction	0	1.0	1.2	0.9	1.3	1.0	1.5	1.3	1.5
Dynamometer testing	0	0.5	1.0	0.6	0.7	0.5	1.3	0.7	1.5
In-vehicle testing	0.6	0.5	1.0	1.5	1.0	0.6	1.8	1.0	1.5
Modifications	0.3	0.5	0.4	0.7	0.5	0.4	0.7	0.5	0.7
Additional testing	0.3	0.5	0.4	0.7	0.5	0.3	0.7	0.5	0.7
Production engineering	0.8	1.0	1.0	1.5	1.0	1.0	1.0	1.0	1.3
Production[b]	2.0	2.0	2.0	2.2	2.0	2.2	2.0	2.0	2.2
Total years of development	4.0	7.0	10.0	9.0	9.0	9.0	12.0	9.0	13.0

[a]Caution: values listed in this table are general. For specific devices, values can differ substantially, and each should be considered on its own merits.

[b]Based on manufacture of 100 units.

TABLE IV-3

Development Time Required for Power-Conversion Devices[a]

(In years)

Phase	Belt-drive system	Torque converter, planetary gear	Hydromechanical power train	Hydromechanical transmission	Hydrostatic (narrow-range) system	Hydrastatic (medium-range) system	Electric-drive dc system	Electric-drive ac system
Research	0	0	0.5	0.5	0	3.5	0.5	4.0
Development design	1.0	1.5	1.5	1.5	1.0	1.4	0.9	2.0
Prototype construction	0.4	1.0	1.5	1.5	1.0	1.0	1.2	1.4
Dynamometer testing	0.4	0.25	0.5	0.5	0.3	0.6	0.3	0.3
In-vehicle testing	0.4	0.5	1.0	1.0	0.9	1.0	1.0	1.4
Modifications	0.3	0.25	0.75	0.75	0.3	0.6	0.4	0.7
Additional testing	0.2	0.5	0.75	0.75	0.3	0.6	0.4	0.5
Production engineering	0.3	1.0	1.5	1.5	0.7	1.3	1.3	1.5
Production[b]	1.0	1.0	2.0	2.0	1.5	2.0	2.0	2.2
Total years of development	4.0	6.0	10.0	10.0	6.0	12.0	8.0	14.0

[a]Caution: values listed in this table are general. For specific devices, values can differ substantially, and each should be considered on its own merits.
[b]Based on manufacture of 100 units.

TABLE IV-4

Probability of Success within Development Periods[a]

Device	Lead time, years	Probability of success
Energy-Conversion Devices		
Reciprocating spark-ignition engine	4	Excellent
Reciprocating compression-ignition engine	7	Excellent
Reciprocating hybrid engine	10	Good
Dynastar engine	9	Good
Rotary-piston engine	9	Good
Stirling-cycle engine	9	Poor
Gas-turbine engine	12	Good
Compound engines	9	Good
Free-piston-turbine engine	13	Good
Power-Conversion Devices		
Belt-drive system	4	Good
Torque-converter planetary-gear power train	6	Excellent
Hydromechanical power train	8	Good
Hydromechanical transmission	8	Good
Hydrostatic (narrow-range) system	6	Good
Hydrostatic (medium-range) system	12	Poor
Electric-drive (dc) system	8	Good
Electric-drive (ac) system	14	Good

[a]Caution: values listed in this table are general. For specific devices, values can differ substantially, and each should be considered on its own merits.

All other types of device would require additional acquisition of tooling and facilities, in various degrees, to meet the US Army's vehicle requirements.

The gas-turbine engine would require the greatest lead time to be produced in quantities needed to meet anticipated requirements. The assumption is made that the US Air Force would hold procurement priority in the area of gas-turbine engines, and the capacity of industry to produce these engines is so limited that difficulty would be encountered in meeting the demands of this one service.

All factors considered in the tradeoff analysis of energy- and power-conversion devices are summarized in Tables IV-5 and IV-6. The devices considered were those believed to hold the greatest potential for improving the physical and performance characteristics of tactical vehicles.

EVALUATION OF ENERGY-CONVERSION DEVICES

In the evaluation of energy-conversion devices it was determined that ammonia-fueled engines, fuel cells, unique energy-conversion devices, nuclear reactors, batteries, the Kuhns gasifier, and steam engines would not offer technological advances that would appreciably improve the physical or performance characteristics of tactical vehicles in the foreseeable future. These devices were not considered in the tradeoff analysis.

TABLE IV-5

Summary of Tradeoff Analysis of Energy-Conversion Devices

Type of device	Hp class	Class quantity	Total quantity	R&D cost, millions of dollars	Unit production cost, dollars	R&D risk	Mobilization capacity, %	R&D time, years
Spark-ignition engine	≤ 120	315,350		2.5	900	Excellent	100	4
	120–250	301,100	616,450	2.5	1,400			
Compression-ignition engine	250–500	77,900		5	8,000	Excellent	90	7
	500–1000	34,450	129,350	12	19,000			
	≥ 1000	17,000		17	24,000			
Hybrid engine	≤ 120	315,350		7	2,100	Good	95	10
	120–250	301,100		9	4,000			
	250–500	77,900	728,800	14	8,000			
	500–1000	34,450		17	16,000			
Dynastar engine	120–250	301,100		4	5,000	Good	80	9
	250–500	77,900	413,450	6	10,500			
	500–1000	34,450		11	23,000			
Rotary-piston engine	≤ 120	315,350		2	450	Good	85	9
	120–250	301,100	616,450	3	750			
Stirling-cycle engine	≤ 120	88,600	88,600	6	1,400	Poor	50	9
Gas-turbine engine	500–1000	34,450		17	19,000	Good	50	12
	≥ 1000	17,000	54,450	25	23,000			
Compound engine	120–250	301,100		12	6,500	Good	70	9
	250–500	77,900	413,450	16	10,000			
	500–1000	34,450		18	20,000			
Free-piston engine	120–250	301,100		12	6,500	Good	70	13
	250–500	77,900	413,450	16	10,000			
	500–1000	34,450		18	20,000			

TABLE IV-6

Summary of Tradeoff Analysis of Power-Conversion Devices

Type of device	Hp class	Class quantity	Total quantity	R&D cost, millions of dollars	Unit production cost, dollars	R&D risk	Mobilization capacity, %	R&D time, years
Mechanical transmission, belt drive	≤120	94,600	94,600	1.75	400	Good	95	4
Torque-converter planetary-gear power train	500–1000	34,100		7	9,500			
	≥1000	17,000	51,100	9	14,500	Excellent	90	6
Hydromechanical power train	≤120	24,500		2	4,000			
	120–250	34,500		3	4,500			
	250–500	76,900	187,000	4	5,000			
	500–1000	34,100		6	6,500	Good	90	10
	≥1000	17,000		9	8,200			
Hydromechanical transmission	120–250	258,200	258,200	1	850	Good	90	10
Hydrostatic (narrow-range) system	≤120	94,600		0.64	2,200	Good	85	6
	120–250	2,500	97,100	0.68	3,500			
Hydrostatic (medium-range) system	≤120	184,650		1	9,800			
	120–250	128,400		2	11,100			
	250–500	1,000	331,400	2	14,760	Poor	75	12
	500–1000	350		2	18,700			
	≥1000	17,000		3	15,800			
Electric-drive (dc) system	≤120	141,600		1	4,800	Good	65	8
	120–250	64,000	205,600	2	8,600			
Electric-drive (ac) system	≤120	90,050		3	8,000			
	120–250	126,000		3	15,000			
	250–500	1,000	234,900	4	35,000	Good	60	14
	500–1000	850		4	50,000			
	≥1000	17,000		5	60,500			

Spark-ignition reciprocating engines will continue to be used in tactical vehicles in power ranges to 250 hp. The spark-ignition engines are relatively compact, have a good power-output-to-weight ratio, and can be procured at a reasonable cost. Industry continues to improve these engines for commercial application, but the engines designed for commercial use must be modified to be acceptable for use in tactical vehicles. There is very little, or no risk, inherent in the successful modification of available commercial engines, and in case of full mobilization, industry could respond rapidly to meet the requirements of the Army.

Compression-ignition reciprocating engines are being developed by industry for commercial application in the power ranges from 100 to 600 hp. These engines require modification to be acceptable for use in tactical vehicles. Since industry is not developing compression-ignition engines above 600 hp the Government must sponsor this development to assure the availability of the engines for tactical vehicles. Compression-ignition reciprocating engines have proved reliable and operate at a lower fuel consumption rate than spark-ignition reciprocating engines but are heavier and cost more to produce. There is very little risk in the successful development of the compression-ignition engines as the design principles have been proved. To improve present-day technology and achieve higher specific compression-ignition engine outputs coupled with a multifuel capability would entail an element of risk. However, the risk is small, and compression-ignition engine development should continue. Should full mobilization be demanded and the development lead time for this engine prove too long to meet requirements, the more conventional compression-ignition reciprocating engine could be produced for an interim period until more advanced engines became available.

The hybrid engine is a comparatively new engine concept that combines the best spark-ignition and compression-ignition reciprocating-engine features. The hybrid engine appears to offer good specific fuel consumption and has multifuel capabilities. The output-to-weight- and -size ratios are better than those offered by either the spark-ignition or compression-ignition reciprocating engines. Although only a few test-stand models have been produced, test results indicate that the hybrid engines have excellent potential for improving the capabilities of many future vehicles. Full development of the hybrid engines would require a long lead time with some element of risk. However, when fully developed these engines could be produced in large quantities, and in relatively short time if required for full mobilization because industries' present facilities and tooling would be applicable. The cost of the hybrid engine in production is estimated to be comparable to that of compression-ignition reciprocating engines.

Gas-turbine engines when developed for tactical vehicles would offer excellent power-to-weight-and-size ratio but cannot compete with reciprocating engines on a unit-cost-per-horsepower basis, except in horsepower ranges from above 900 to 1000 hp. Gas-turbine engines, considering the life expectancy and reduced maintenance of these engines, are competitive with reciprocating engines in horsepower ranges as low as 500 to 600 hp. The cost for developing gas-turbine engines is relatively high considering the low density of the vehicles using this engine plus the element of risk involved in the development of this engine. In addition, in case of full mobilization, the limited production quantities of these units would be absorbed by the Air Force, whose priority could demand most of the units produced. In the power ranges above 1000 hp, the development of the gas-turbine engine should continue since there are no engines of other types for tactical vehicles (such as the main battle tank) that could easily meet the space and weight limitations imposed and remain in the required power range. Should a successful gas-turbine engine in a power class over 1000 hp be developed and produced, consideration could be given to the development of the same type of engine in the 500- to 1000-hp class.

Rotary-piston engines now are being developed and funded by industry for commercial vehicle application. Industry has produced some prototype engines in power ranges up to 250 hp that appear to offer a greater specific power output for weight and size than any other engine. The design is simple; therefore the unit cost of this engine in production quantities should be much lower than that of present-day reciprocating engines. This type of engine, more than any other, lends itself to the "throwaway" item category in case of major repair or overhaul. The Curtiss-Wright Corporation has been most successful in developing the rotary-piston engine. One major problem is that Curtiss-Wright has proprietary rights on the engine, and the capacity of industry to produce these engines at a competitive cost will depend on successful negotiations with the Curtiss-Wright Corporation. Present prototype engines are applicable only to commercial vehicles. Development work would be required to produce reliable tactical-type vehicle engines.

The Dynastar engine is a reciprocating engine with peripheral opposed pistons that offers more compactness than the more conventional reciprocating engines. Only a few prototype engines have been produced. One was successful when tested in a small Coast Guard vessel. The costs of developing the Dynastar engine for use in tactical vehicles are estimated to be nearly the same as compression-ignition reciprocating-engine development costs. There is some doubt that the Dynastar engines, when developed, would offer any improvements over the newer compression-ignition engines. Industries' production capacity for the Dynastar during full mobilization would be limited since neither facilities nor production tooling are available.

Basically the compound engine is a compression-ignition reciprocating engine that incorporates a power turbine driven by the engine's exhaust gases. Used for aircraft, the compound engine has not been installed in vehicles furnishing ground power. Compound engines developed for tactical vehicles would be applicable in power ranges from 120 to 1000 hp. The compound engine has a lower fuel consumption rate for power output than any other engine considered; however, the development and unit costs would be somewhat high owing to engine complexity. There is some element of risk in the development of the compound engine for tactical vehicle use. However, if maximum fuel economy is of prime concern, development of this engine would be warranted. Industry's capacity to produce compound engines during a period of full mobilization would be limited because gas-turbine engine components would be difficult to obtain.

Free-piston engines have been developed for application in a few commercial vehicles with limited success. The free-piston engine offers excellent torque characteristics, has a low fuel consumption rate, and multifuel capabilities. The greater weight and size of this engine compared with the power output does limit its use in many tactical vehicles so that a choice of engines based on weight and size would favor the conventional engines. Considerable research is required for the full development of the free-piston engine, and a long lead time would be required to produce the engine in production quantities because neither facilities nor tooling appropriate for the task are available. The development and unit cost in production of these engines for tactical vehicles is somewhat high when compared to the production costs of conventional reciprocating engines. The free-piston engine, if developed with success, would offer good fuel consumption characteristics and a multifuel capability to offset its additional engine size and weight.

A few Stirling-cycle-engine prototypes that operate at low noise levels have been developed in the lower power ranges. The latest Stirling-cycle-engine development, called the "Dineen" process, shows promise to increase the engine output-to-size-and-weight ratio. However, even in the low power ranges to 120 hp, inasmuch as these engines are quite complex, successful development would entail a large degree of risk. The production costs of the Stirling-cycle engine would be higher than those of the more conventional engines. The density of use for this engine in tactical vehicles is relatively low, and the complexity of the engine is high. Only limited quantities could be produced in a period of full mobilization since facilities and tooling for such production would be limited. However, an effort should be made to develop this silently operating engine, because silent operation is not available in any other acceptable engine.

EVALUATION OF POWER-CONVERSION DEVICES

Various types of power-conversion devices were evaluated to determine which devices had the greatest potential for contributing most, if fully developed and available, to tactical-vehicle capability. In the course of this evaluation the mechanical power-conversion devices found unsuitable for use in tactical vehicles were the progressive-sliding-gear, selective-sliding-gear, and constant-mesh transmissions. Of the hydrokinetic power-conversion devices, the "Hydramatic," the torque converter, and Torqmatic devices were found unsuitable.

The synchromesh transmission has application for tactical vehicles of small horsepower requirement. Industry has and will continue to develop synchromesh transmissions to meet specific tactical-vehicle requirements. The cost of the units is relatively low, and full production in a period of mobilization would require minimal lead time.

Mechanical belt-drive transmissions have been developed and used with success (by industry) in low-powered commercial vehicles. However, these transmissions require additional development to assure reliable performance when used in tactical vehicles that operate under severe environmental conditions. The mechanical belt-drive transmissions are quite simple, have automatic provision for varying speed torque ratios, and can be produced at a low unit cost. However, the power range is limited to 120 hp. There is little risk in the development of belt-drive transmissions because the principles have been proved. A development program directed to improvement of unit reliability in tactical vehicles could be accomplished within a relatively short lead time. The capacity of industry to produce the mechanical belt-drive transmission is good because many of the components used are similar to components now available. The primary advantage of using this type of transmission in tactical vehicles would be low unit cost for an engine in a restricted power range.

Torque-converter planetary-gear power trains have been used with success in commercial vehicles as well as military tactical vehicles. The X series represents the latest and most advanced military version of this type of power train, but it is not available in the power ranges over 600 hp. The X series power train is compact, provides for ease of operation with excellent vehicle mobility, and has higher specific outputs than other available hydrokinetic units. The full development of the X series power train in all power ranges would improve the capabilities of tactical vehicles. Such development could be achieved in a relatively short time frame. The element of risk for successful development of the X series in the higher power ranges is small because units of this series have been proved in the lower power ranges. Unit costs in production quantities also would be proportional to unit power output. Industry's capacity to produce these power trains at time of full mobilization would be excellent since production facilities are available and much special tooling now available would be applicable for use.

The concept of the hydromechanical power-conversion device is relatively new. A few prototypes have been installed in both wheeled and tracked vehicles. The prototypes produced are in power ranges to 260 hp. Test results indicate that hydromechanical power-conversion devices have a potential of improving the capabilities of most tactical vehicles as they offer higher output for unit weight and size and incorporate fewer parts than present hydrokinetic power-conversion devices. One excellent feature of this device is its capability to enable engine operation at optimum speed-power range independent of vehicle speeds. The most economical fuel consumption results.

A hydromechanical power train is basically made up of two symmetrically opposite hydromechanical transmissions. Consequently, power-train development will benefit from the development of these transmissions. Development of the devices will require a long lead time with some inherent element of risk for success. However, until the more advanced hydromechanical units become available, hydrokinetic power-conversion devices are available for use in tactical vehicles.

It is estimated that the development costs of hydromechanical power-conversion devices are comparable to other previously developed power-conversion devices. However, the unit cost in production quantities would be appreciably lower. In a period of full mobilization these devices could be produced (in the required quantities) in a relatively short time. Most facilities in current use and much of the present tooling could be utilized.

Hydrostatic power-conversion devices are used in vehicles where vehicle space and configuration do not readily permit the use of more conventional devices. The application of hydrostatic devices (in the narrow speed-torque range) proved successful in many slow-moving vehicles. Devices in the medium speed-torque range have had a marginal degree of success in some special types of tactical vehicle. The hydrostatic drive in the narrow speed-torque range has application for special tactical vehicles in power ranges to 250 hp and in the medium speed-torque range to 1500 hp. There is a considerable element of risk in the achievement of successful development of these drives in the medium speed-torque range owing to the difficulty of obtaining unit efficiency throughout this wider range.

The cost to develop medium speed-torque-range hydrostatic drives appears to be reasonable, but the estimated unit cost of these devices in production quantities appears to be higher than that of most other conventional power-conversion devices. In a period of full mobilization, industry has the capacity to produce narrow speed-torque range hydrostatic drives since many of the components are available. However, industry lacks the facilities and special tooling needed for quantity production of the medium speed-torque range drives. Hydrostatic drives are applicable for special tactical vehicles (where the more conventional power-conversion devices cannot be readily installed) and allow greater vehicle design flexibility.

Electric-drive (dc) systems have been developed and installed in many commercial vehicles and have performed with success in the low-powered vehicles. These drives now are finding acceptance in the larger high-powered commercial vehicles. The dc drives offer power to all wheels and are ideal for use where vehicle configuration does not readily permit the use of more conventional power-conversion devices. However, as tactical vehicles operate under more severe environmental conditions than do commercial vehicles, the dc electric drives must be modified for improved reliability before they are suitable for use in tactical vehicles. The cost to develop dc drives for tactical-vehicle use would be relatively low since many of the drive components have been proved and many only require modification. Conversely the unit costs of the dc drives would be relatively high since the system is more complex and requires many more components than do the more conventional power-conversion devices. The lead time to develop the dc drive would be short and entails little risk for success. The capacity of industry to produce the devices during full mobilization would be somewhat limited since sufficient production facilities and tooling would not be available.

A recent concept advocates the design of the electric-drive system to operate on alternating current. The ac system seems to offer a reduction of weight and size as compared to a dc electric-drive system. A few prototype systems have been installed in vehicles for the purpose of obtaining data, but none of the present ac electric-drive system installations can be considered acceptable. An ac electric-drive system offers features that could provide auxiliary power and permit optimum engine operation at speeds in the most economical fuel consumption range independent of vehicle speeds. This type of system could be developed for tactical vehicles in power ranges up to 1500 hp. Since there are very few commercial vehicular applications for an ac system, industry has no incentive to develop the ac drives. Research and development costs would be moderate, but the estimated unit cost is higher than that of the more conventional power-conversion devices. The system's cost is high because of the cost of some of the quite complex components. There is an element of risk in bringing this ac electric-drive system to full development because the reliability of many of the components remains to be proved, and it may be difficult to reduce component weight and size. Successful development of the system would require a long lead time and in the case of full mobilization production would be limited since special tooling and facilities would not be readily available.

Chapter 25

FINDINGS, CONCLUSIONS, AND RECOMMENDATIONS

FINDINGS AND CONCLUSIONS

Energy-Conversion Devices

Numerous energy-conversion devices were considered in the course of the review and evaluated to enable selection of devices that warrant further R&D by the Army. A brief discussion of these devices has been presented. Fourteen energy-conversion devices, plus variations of certain types, were given close scrutiny. The pertinent findings related to these 14 devices are given in the following paragraphs.

Ammonia-Fueled Engines. US Army-sponsored R&D of ammonia-fueled engines for use in tactical vehicles is not warranted at this time. In combat, ammonia engines would be highly vulnerable, necessitating two types of vehicle for the same horsepower class. One type would be required for combat and a second for logistics. Introduction of ammonia engines to the military supply system would immediately impose two new logistic problems. (1) The number of vehicles needed would increase because the ammonia engine cannot be used under combat conditions. This situation nullifies the advantage of a decreased hydrocarbon-fuel transportation burden. (2) An entire new line of spare parts would be required in the military logistic system. In addition, supply depots would have to accommodate and implement issue of the new ammonia-fueled engines, and the maintenance depots would have to gear for their repair.

Fuel Cells. Army-sponsored R&D on fuel cells is not warranted at this time because this device will remain too bulky, heavy, and inefficient within the foreseeable future for use in tactical vehicles. R&D of fuel cells should be held in abeyance until successful application is attained in stationary equipment. If, in the future, fuel cells do come into use in stationary equipment and if many of the present problems are resolved, R&D of these cells for tactical-vehicle applications should be reconsidered.

Unique Energy-Conversion Devices. Army-sponsored R&D of unique energy-conversion devices is not warranted at this time. Nine such devices considered in the course of this study were found to be much too heavy, to require too much space, and to be too inefficient for application in tactical vehicles. Development of unique devices within the foreseeable future would not result in an energy-conversion device that would improve the capability of tactical vehicles.

Nuclear Reactors. Army-sponsored R&D of nuclear reactors is not warranted at this time because they are, and will remain, too bulky and too heavy for application in tactical vehicles. Further, materials to provide efficient shielding must be developed and a concept devised to overcome personnel and vehicle vulnerability to radiation hazards. In addition the costs of such a device require substantial reduction.

Batteries. Army-sponsored R&D of batteries should be limited to those applications in tactical vehicles that have a restricted operating range and those few specific applications in which a battery energy device is most appropriate and silent operation is mandatory. For most other tactical-vehicle operations, the operating range provided by battery energy devices is too limited, and the device is too heavy to warrant development.

KGG Cycle or "Kuhns" Gasifier. Army-sponsored R&D of the "Kuhns" gasifier-turbine engine is not warranted until a thorough cycle and mechanical analysis is conducted, or the successful development of this device is first achieved for use in aircraft. At that time, if feasible, additional development should be considered for tactical-vehicle application.

Steam Engines. Army-sponsored R&D of steam engines is not warranted. To date a concept has not been devised that would reduce the weight and size of a steam engine below that of present-day conventional reciprocating engines. Further consideration should not be given the steam engine unless the development of a vehicle is desired in which weight and size are relatively unimportant, high torque characteristics are required at low power output, and silent operation of this vehicle is considered to be of utmost importance.

Reciprocating Piston Engines. Army-sponsored R&D of reciprocating piston engines is warranted.

Spark-ignition reciprocating engines are being developed by industry for commercial application, but these engines must be modified before they can be used in tactical vehicles.

Compression-ignition engines are being developed by industry in power ranges up to 600 hp for commercial application, but these engines must also be modified before they can be used in tactical vehicles. In addition, development programs are required for engines in power ranges over 500 hp.

Hybrid engines combine the best features of the spark-ignition and compression-ignition reciprocating engines and appear to have the best potential for improving tactical-vehicle capabilities. Development of these engines in most power ranges should be conducted; however, the development will require a long lead time.

Gas-Turbine Engines. Army-sponsored R&D of the gas-turbine engine is warranted. The gas-turbine engines could have practical application in vehicles where high power is required and where vehicle space and weight for engine accommodation are limited.

Rotary-Piston Engines. Army-sponsored R&D of the rotary-piston engine is warranted. Curtiss-Wright Corporation's "Wankel" rotary-piston engine shows promise of improving the physical and performance characteristics of many future tactical vehicles. Successful development could result in an engine of fewer parts that offers a greater horsepower output for engine size and weight than currently available engines. Further, tactical-vehicle power-producing components could be discarded rather than repaired.

Dynastar Engines. Army-sponsored R&D of the Dynastar engine is warranted. Thiokol Chemical Corporation's Dynastar engine shows promise to produce a spark-ignition or a compression-ignition engine that could improve the physical and performance characteristics of many future tactical vehicles. Dynastar development would result in an engine that has high-horsepower-output-to-size and high-horsepower-output-to-weight ratios and multifuel capabilities. The Dynastar concept leads itself to design applications that may use a family of engines having many common parts.

Compound Engines. Army-sponsored R&D of the compound engine is warranted. Compound engines could have practical application in certain vehicles where low fuel consumption is of greater importance than size, weight, and engine complexity. Compound-engine development should not be sponsored for the purpose of using these engines as gas generators to supply gas to a free-turbine power output.

Free-Piston-Turbine Engines. Army-sponsored R&D of the free-piston-turbine engine is warranted. These engines could have practical application for some future tactical vehicles where long periods of continuous operation are required and where low fuel consumption coupled with a multifuel capability is of prime importance.

Stirling-Cycle Engines. Army-sponsored R&D of the Stirling-cycle engine is warranted. A concept utilizing the "Dineen" process appears promising and could lessen present engine size and weight. A Dineen-process Stirling-cycle engine would be capable of operation at noise and vibration levels that are much lower than those of a gasoline or diesel reciprocating engine. This engine would be capable of operation on various hydrocarbon fuels. Should low efficiencies that cannot be resolved be encountered at any time during engine development, the value of further research could then be determined. The Stirling-cycle engine does show promise for application in tactical vehicles where silent operation is required.

Power-Conversion Devices

Mechanical Power-Conversion Devices. Mechanical power-conversion devices have been used with success over a period of many years in tactical as well as commercial wheeled and tracked vehicles. Today most mechanical power-conversion devices have outlived their usefulness for tactical-vehicle application. An exception is the synchromesh device that will continue to have application in low-powered vehicles (to 120 hp). Industry is continually developing and improving synchromesh devices for commercial vehicles, and these can be readily modified for use in tactical vehicles.

Since tactical-vehicle quantities are low compared to those of commercial vehicles, the cost of synchromesh devices produced specifically for tactical vehicles would be higher than those produced for commercial vehicles. Also it is most probable that any units produced for a broad class of tactical vehicles would require some modifications prior to their acceptance in a specific vehicle. Therefore Army-sponsored R&D of synchromesh power-conversion devices is not warranted.

The belt drive is another mechanical power-conversion device that has been investigated. Belt-drive units are used in commercial vehicles but have not as yet been proved acceptable for use in tactical vehicles. This device appears to have the potential for successful application in some types of small wheeled tactical vehicles up to 50 hp. A belt drive can be produced at a low unit cost and would provide ease of operation at relatively slow speeds. Army-sponsored R&D of belt-drive transmissions for use in tactical vehicles is warranted. The development of belt-drive transmissions should, however, be confined to those areas where low cost is of prime importance.

Hydrokinetic Transmissions and Power Trains. Army-sponsored R&D of the Hydramatic and Torqmatic transmission is not warranted. Both the Hydramatic and Torqmatic transmissions (hydrokinetic units) are available in the commercial market. Although each of these transmission units is applicable to commercial vehicles, only the Torqmatic transmission has applications for use in military tactical vehicles, and these applications are restricted to a limited number of kinds of tactical vehicles.

The torque-converter planetary-gear transmissions are used with success in tactical wheeled vehicles, as transmissions, and as a segment of the power train in tracked vehicles. Torque-converter planetary-gear transmissions are available in all required torque ranges for use as a transmission or a power train segment in both wheeled and tracked vehicles.

The X series power train, a series of torque-converter planetary-gear transmission of the hydrokinetic family, was found to provide superior tracked vehicle performance. Although the X series power train provides tracked vehicles with a performance capability superior to that provided by any other hydrokinetic power train, the X series is available only in torque ranges of from 200 to 750 lb-ft. Therefore R&D is required to extend the range of the X series hydrokinetic power train and furnish some improvement to the present units. Though the benefits derived from a fully developed hydromechanical transmission and power train may well exceed those furnished by a fully developed hydrokinetic X series power train, the element of risk attendant on development of the X series is less. Full development of the X series assures an advanced power train in the high power class until the time that hydromechanical transmissions are developed and proved acceptable.

Hydromechanical Transmissions and Power Trains. Army-sponsored R&D of hydromechanical transmissions and power trains is warranted. As a hydromechanical power train is basically made up of two symmetrically opposite hydromechanical transmissions, the development of the power trains will benefit the transmission development.

Hydromechanical power trains appear to have a greater potential for improving the capabilities of tactical tracked vehicles than all other power trains evaluated. The units could offer improved specific-weight- and volume-to-torque ratios, better acceleration, and greater fuel economy. However, the full development of hydromechanical transmissions and power trains in all torque ranges would require at least 10 years.

Industry is engaged in the R&D of hydromechanical power-conversion devices for commercial application. The effort is limited for the most part to keeping abreast of advanced technology rather than to full development of the hydromechanical devices. The impact of the technology is less on commercial vehicles than on tactical vehicles. Industry has little incentive to fund development for hydromechanical devices since the commercial market is limited for the high expenditure of funds demanded. Tactical vehicles operate in a much more severe environment and could take full advantage of the advanced technology. Therefore the achievement of a full range of hydromechanical power-conversion devices will depend on Government support for development. The successful development of hydromechanical power trains would result in a unit that could be readily applied to transmissions, and the components would be common to both wheeled and tracked vehicles. Thus the logistic burden would be reduced.

Hydrostatic Drives. Army-sponsored R&D of hydrostatic drives for tactical vehicles is warranted. Narrow-range hydrostatic drives are being developed by industry for slow-moving commercial vehicles. There is little incentive for industry to develop medium-range hydrostatic drives since the application to commercial vehicles is very limited. The military has a need for both the narrow- and medium-range hydrostatic drives for special tactical vehicles. Therefore Army-sponsored R&D are required to modify commercially available narrow-range hydrostatic drives to improve drive reliability and to produce satisfactory medium-range hydrostatic drives for tactical vehicles.

Dc and Ac Electric-Drive Systems. Army-sponsored R&D of electric-drive systems is warranted. Both the dc and ac electric-drive systems improve the physical and performance characteristics of some types of tactical vehicles. The review performed as part of this study indicates that greater success has been experienced in the development of dc electric-drive systems than has been experienced with ac systems up to the time of writing. On the basis of this finding, the most opportune course to pursue would be to develop dc electric-drive systems and at the same time to conduct R&D of ac. The dc electric-drive system could be available for tactical vehicles within a much shorter time frame than the ac electric-drive system.

Electric-drive dc systems are being developed by industry for commercial vehicles not required to operate under environmental conditions as severe as those to which tactical vehicles are exposed. The use of dc drives in tactical vehicles is of particular value in applications in which the configuration of vehicles does not permit the installation of more conventional power-conversion devices. However, commercial dc electric drives must be modified to be acceptable for tactical vehicles. Therefore, to provide for such modification programs, US Army support is required.

Electric-drive ac systems are not being developed by industry since the application of an ac drive for commercial vehicles is very limited. Military vehicles require a wider speed-torque range than most commercial vehicles and therefore would have a greater application for ac electric-drive systems. To develop the potential that ac electric drives could offer would require US Army support.

RECOMMENDATIONS

The recommendation is made that the US Army sponsor R&D programs for the energy- and power-conversion devices listed below. The order of priority is as indicated. Suggestions for specific areas of device development and methods of implementation are offered.

Energy-Conversion Devices

The energy-conversion devices listed below, in order of priority, are selected as those that will, when fully developed, make the greatest contribution toward satisfying the energy-conversion-device requirements of military tactical vehicles. The evaluation and final review of this study predicts a greater "across-the-board" benefit from the first four devices listed than is anticipated from all other devices recommended or considered. Further, these four engines represent the minimum number of such devices that should be developed. Benefit would derive from the development of all devices recommended but the benefits derived from the latter five will have the lesser impact on the technology.

1. Spark-Ignition Reciprocating Engines. Continued improvement of spark-ignition reciprocating engines for tactical use should be made through modification of available commercial engines.
2. Compression-Ignition Engines. Available commercial compression-ignition engines should be improved in the power ranges under 500 hp. New engines for military use in power ranges over 500 hp should be developed.
3. Hybrid Engines. Hybrid engines should be developed in power ranges of 1000 hp and below.
4. Gas-Turbine Engines. Gas-turbine engines should be developed in power ranges over 1000 hp.
5. Rotary-Piston Engines. Rotary-piston engines should be developed in power ranges to 250 hp.
6. Dynastar Engines. Dynastar engines in the 120- to 250-hp range should be developed.
7. Compound Engines. Compound engines in the power ranges between 250 to 1000 hp should be developed.
8. Free-Piston Engines. Free-piston engines should be developed in the power range between 120 and 1000 hp.
9. Stirling-Cycle Engines. Stirling-cycle engines should be developed in those low power ranges under 120 hp.

Power-Conversion Devices

The seven power-conversion devices listed below are selected as those that will when fully developed make the greatest contribution toward satisfying the requirements of military tactical vehicles. An order of priority has not been established for the power-conversion devices recommended since many types of vehicle will undergo concurrent development. The application of power-conversion devices depends in large measure on the physical and performance requirements of the vehicle.

Torque-Converter Planetary-Gear Power Trains. Torque-converter planetary-gear power trains should be developed in power ranges about 600 hp.

Hydromechanical Power-Conversion Devices. Hydromechanical power-conversion devices should be developed in all power ranges.

Ac Electric-Drive Systems. Electric-drive ac systems should be developed in all power ranges.

Hydrostatic-Drive Systems (Medium Range). Hydrostatic-drive systems in the medium range should be developed in power ranges to 1000 hp.

Hydrostatic-Drive System (Narrow Range). Hydrostatic-drive systems in the narrow range should be developed in the power ranges under 250 hp.

Dc Electric-Drive Systems. Electric-drive dc systems should be developed in the power ranges under 250 hp.

Belt-Drive Transmissions. Belt-drive transmissions should be developed in power ranges under 120 hp.

Component Standardization, Logistics, and Cost Considerations

Although the need for component standardization with the attendant logistics and cost considerations has been recognized and given full consideration throughout this study, it is recommended that caution be exercised against the adoption of a too rigid standardization policy. Goals for standardization should not be so firm as to impede the acceptance of new developments, which impedance would in turn retard improvements in technology.